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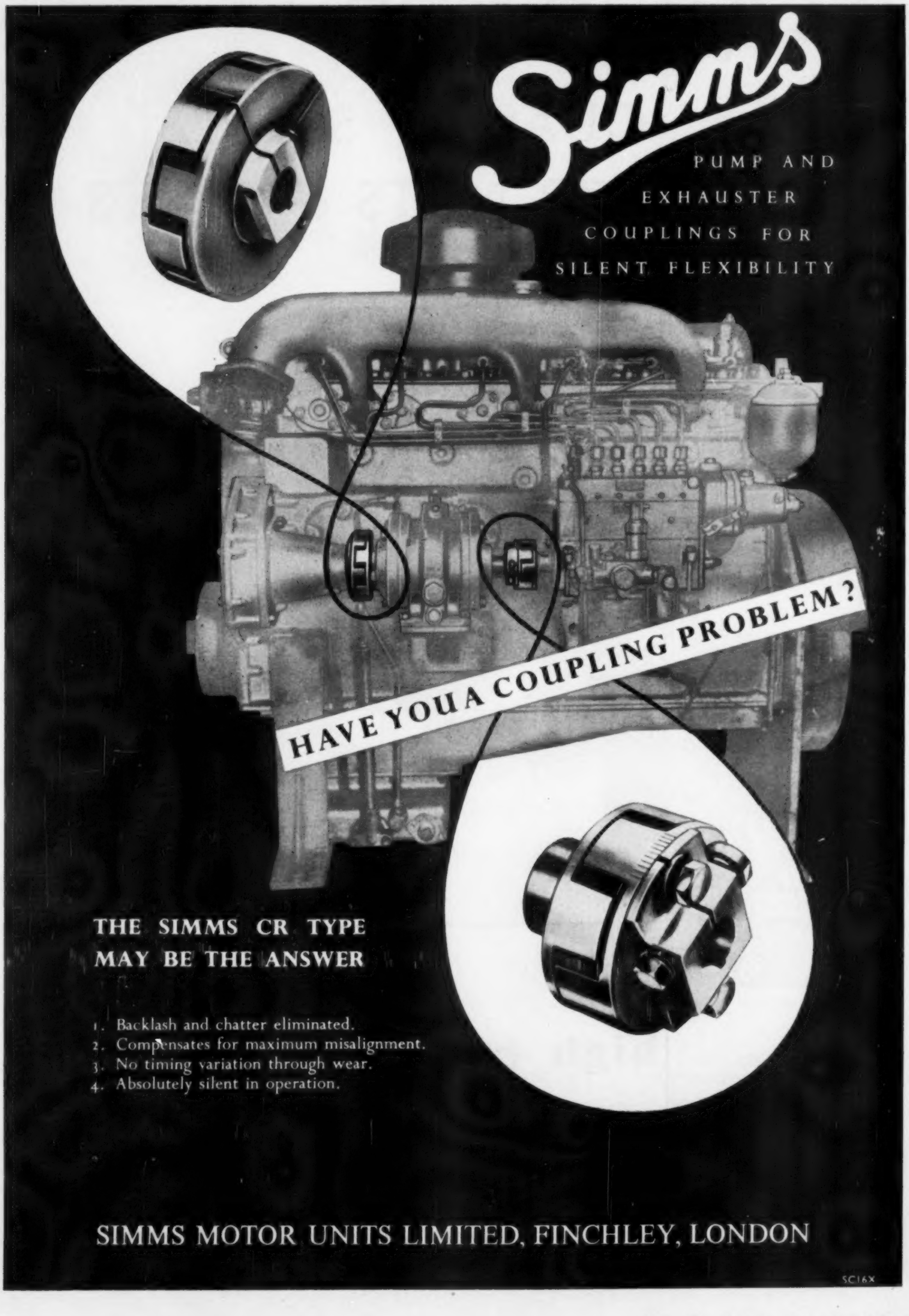
Vol. 48 No. 13 DECEMBER 1958 PRICE: 3s. 6d.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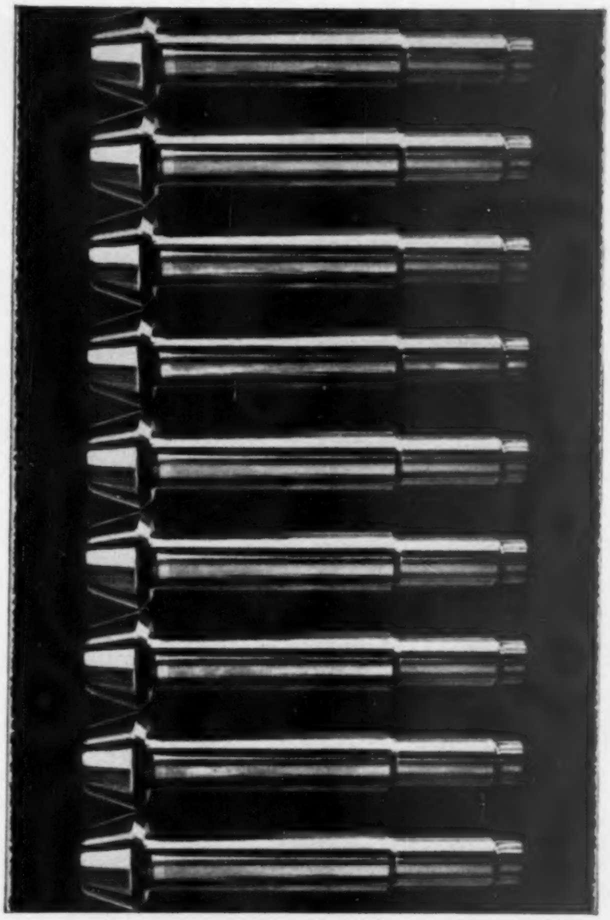

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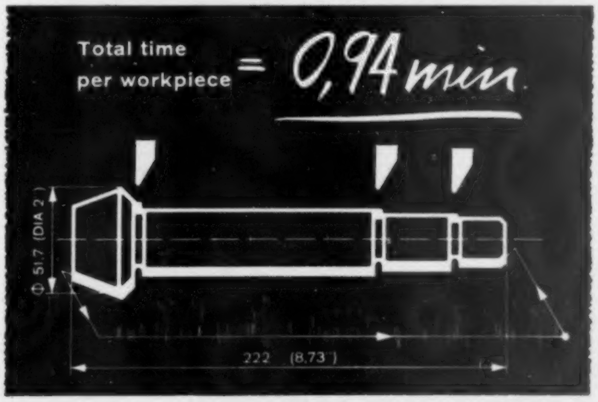

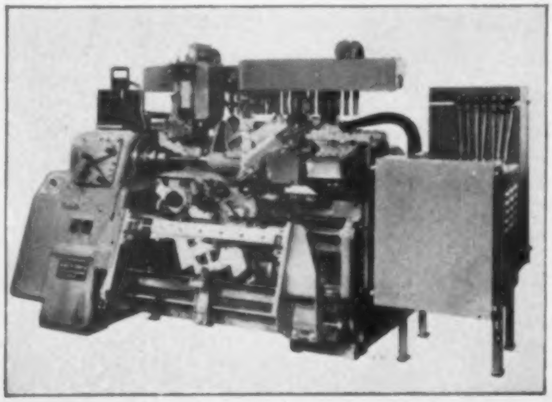

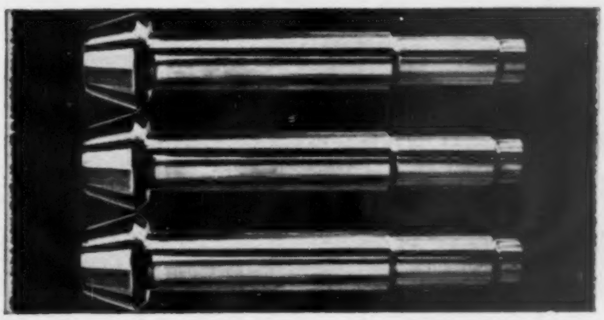

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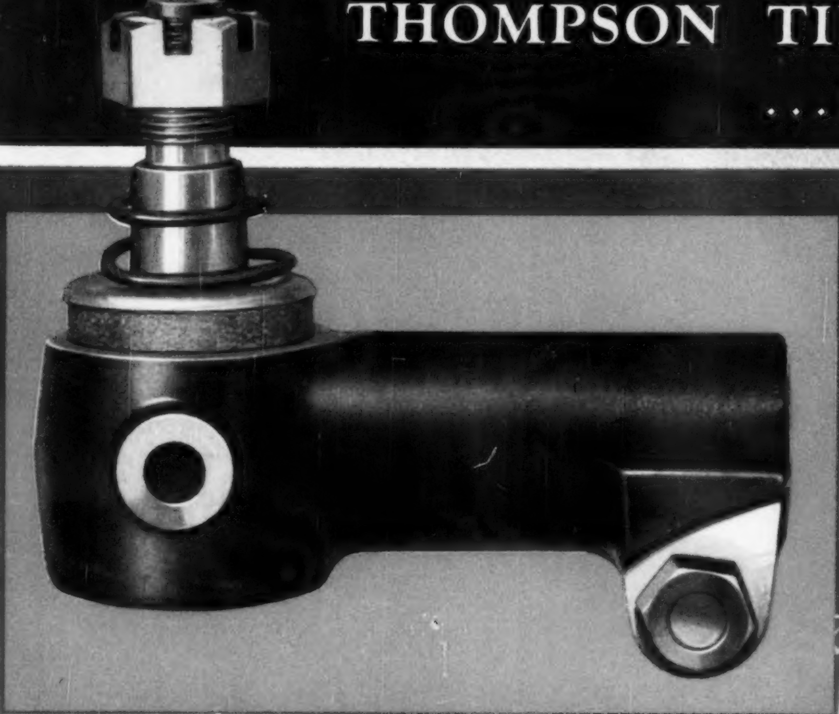

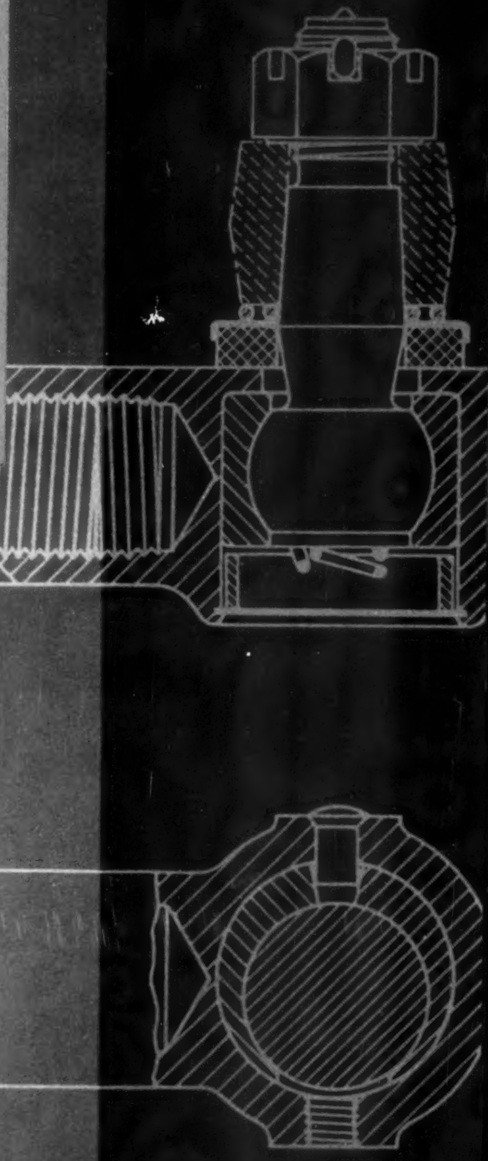

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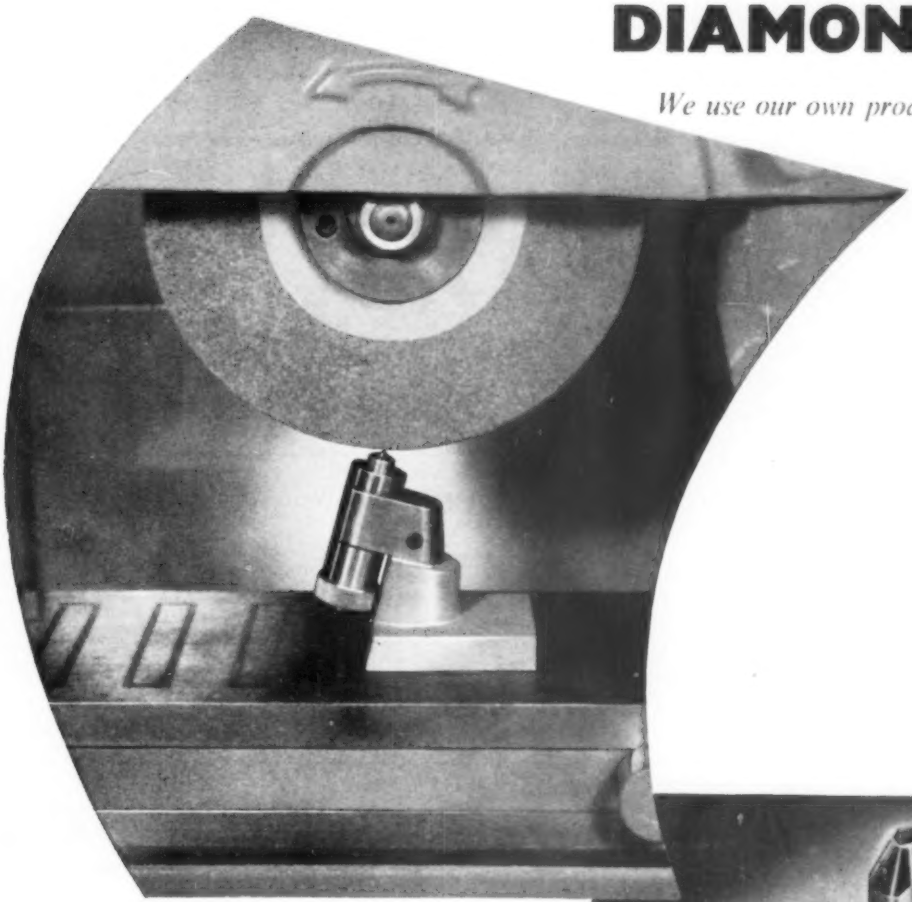

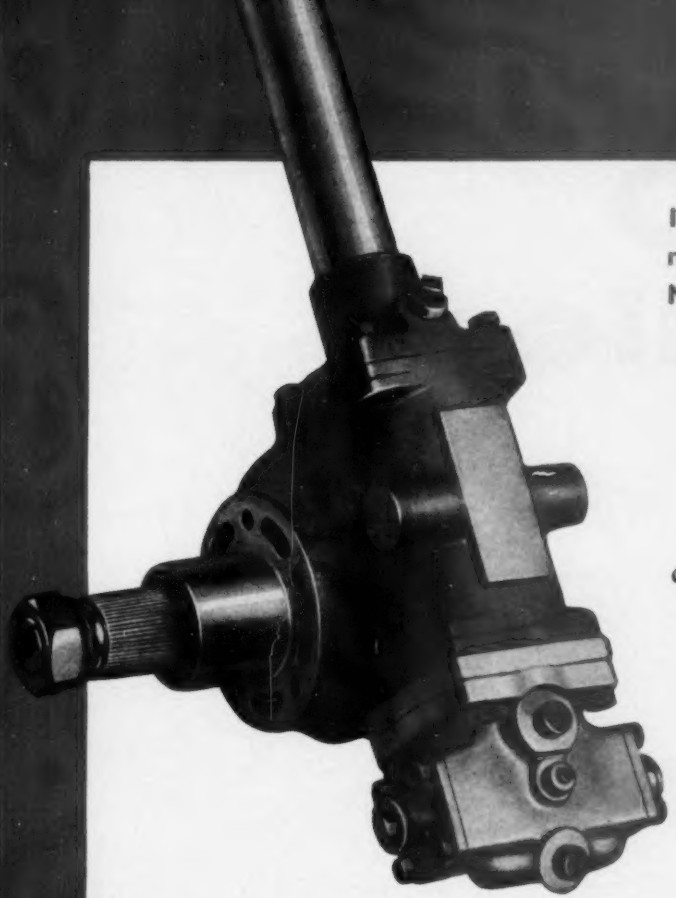

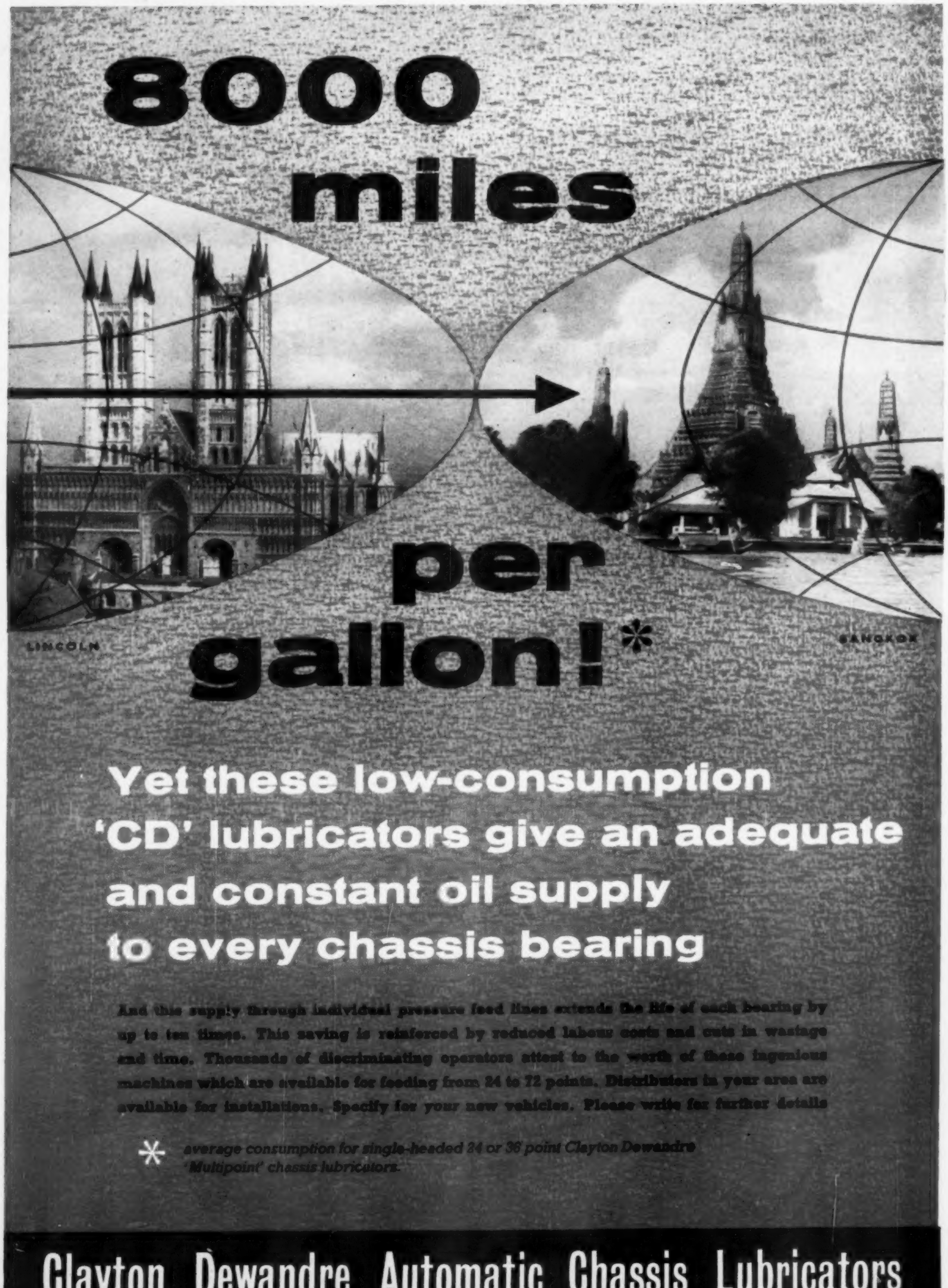
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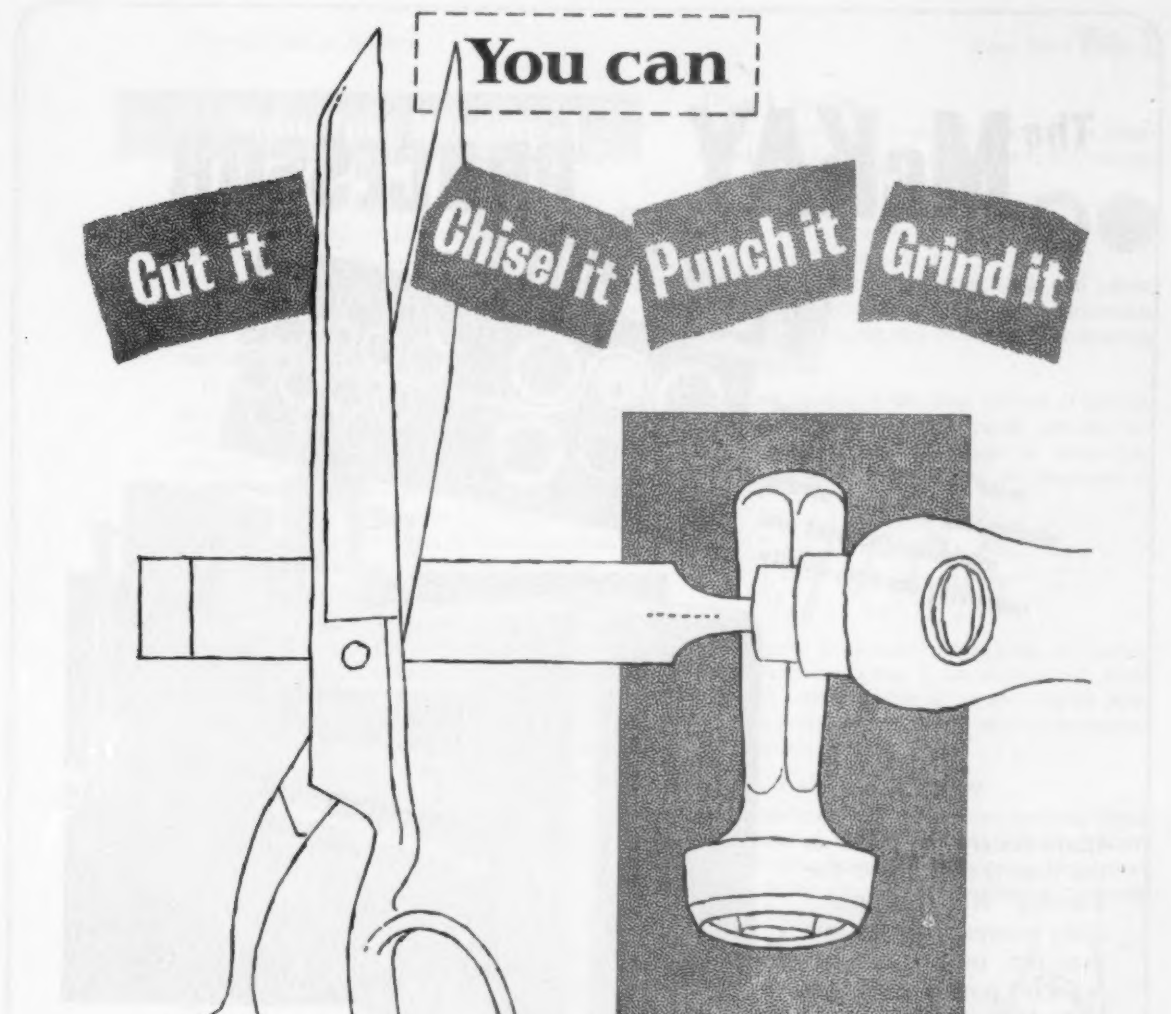
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Electrical Aids in Industry | *Data Sheet* **No. 2**

Induction Heating-1

Electro-magnetic induction was discovered by Faraday and used by him to evolve the electric motor. In some cases, induction can be a nuisance by causing heat losses to arise from induced eddy currents in nearby pieces of metal, but in industry today those heat losses are being turned to good account as a method of metal heating.

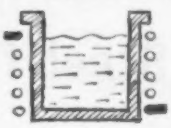
Induction heating produces heat only in the work piece. This is a fascinating and spectacular process which has the additional merit of being extremely effective. An important feature of induction heating is that it gives rapid temperature rise with no time lag, starting being instantaneous. Perhaps its greatest advantages are that it can be used in automatic processes and does not demand skilled labour.

Induction heating can deal efficiently with all the applications outlined below, and including metal melting, preheating and stress relieving, surface hardening, heating of large components, preheating of steel tubes for manipulating, heating for shrinking, for forging and extrusion, as well as for the heating of vessels. There are various methods of applying it, the method to be used being dictated by the application.

Technical details relating to the choice and use of individual types of induction heating will be set out in a subsequent data sheet.

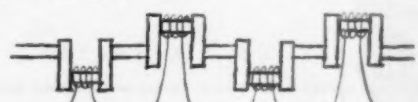
Metal melting

The oldest and largest application of induction heating lies in the melting of metals. The outstanding advantages are: freedom from deleterious gases and products of combustion and other contaminants, rapid speed of melting, low running costs, improved working conditions and reduced metal loss.



Metallurgical processes

This covers a large variety of processes which may be basically divided into surface heating and through heating. Most processes fall into the former category and are used for the purpose of



skin hardening. They include the treatment of such components as: pins, camshafts, crankshafts, rollers, cylinder liners, gear teeth, rocker arms and shafts and valve stems.

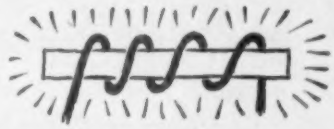
5857

Through Heating

When the current in an induction coil is maintained for a longer period it results in the heat engendered in the work piece penetrating to a greater depth and eventually throughout the piece. It is largely used for the heating of slugs and billets, for upsetting and upset forging, flanging and nosing, shearing and forging and bolt blanks for heading.

Annealing

Induction heating is the ideal method of altering the character of metals for a special purpose. Annealing and its related process of tempering, normalising and stress relieving are prominent in



this category. In the case of annealing, one feature of the treatment is that it can be localised, while induction tempering can do in one minute what would require half-an-hour or more with conventional furnace heating.

Brazing, soldering and welding

Brazing and soldering by induction are the quickest and cleanest methods of joining metals together, and it is often beneficial to re-design the parts so that the fullest advantage can be taken of induction heating. Brazing can be used in the case of: carbide inserts for tools, tips for rock drills, parts for universal joints, etc., while the many applications of soldering are well-known. It was not until well into the present century that the age-old craft of welding became a positive aid to greater productivity. Induction welding produces a good, clean weld because oxidation is almost absent.

Miscellaneous Application

Induction heating can be profitably used for a large variety of purposes, including the fusion and hot pressing of powders, heating *in vacuo*, gas determination in metals, fusion of glass, chemical work and many other processes.

For further information, get in touch with your Electricity Board or write direct to the Electrical Development Association.

Excellent reference books are available on electricity and productivity (8/6 each or 9/- post free)—"Induction and Dielectric Heating" is an example.

E.D.A. also have available on free loan a series of films on the industrial use of electricity. Ask for a catalogue.

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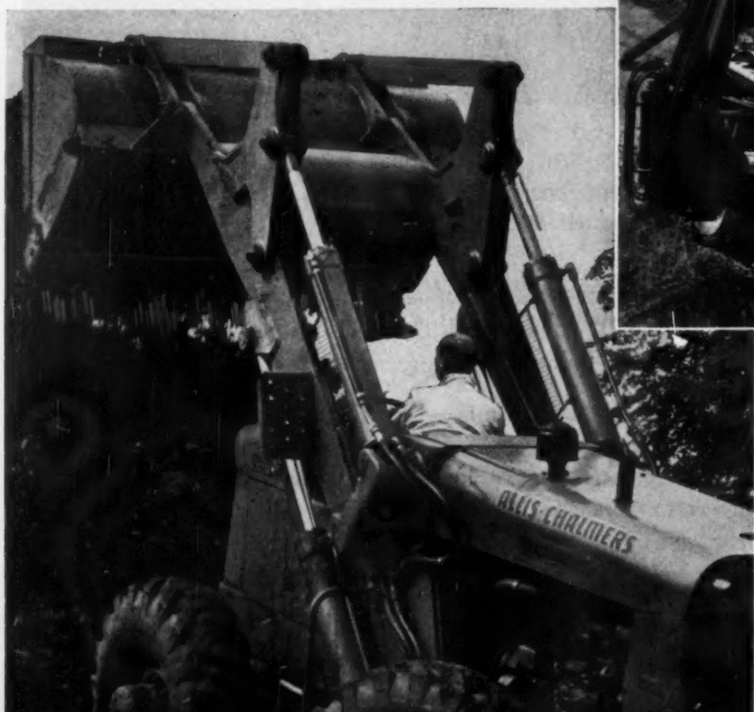

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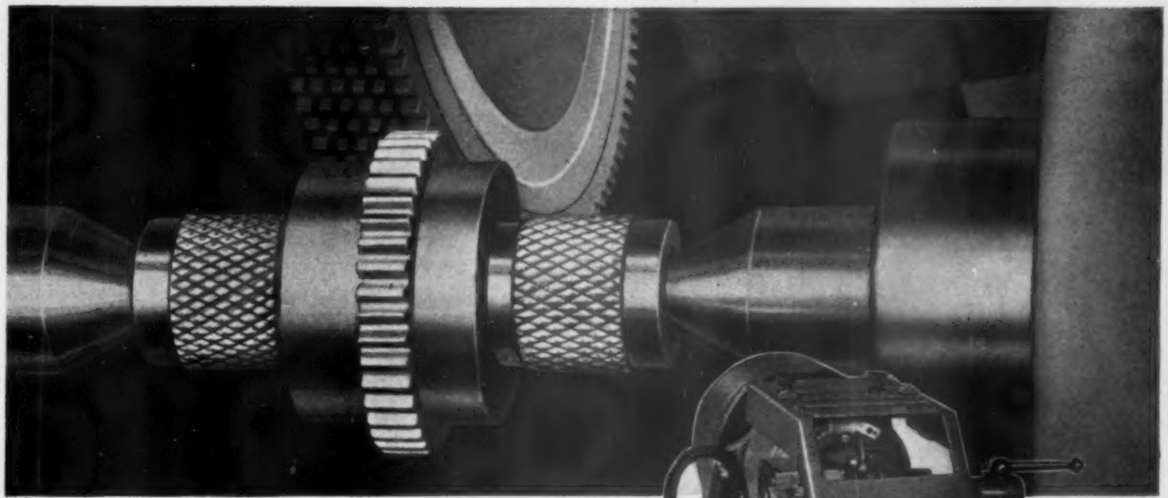


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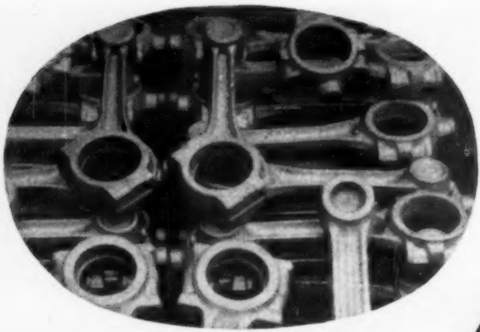

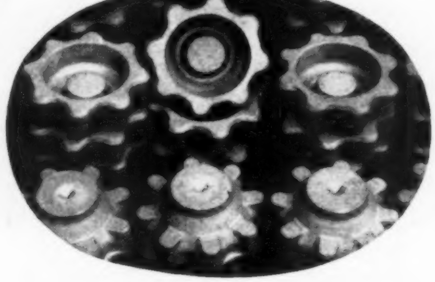

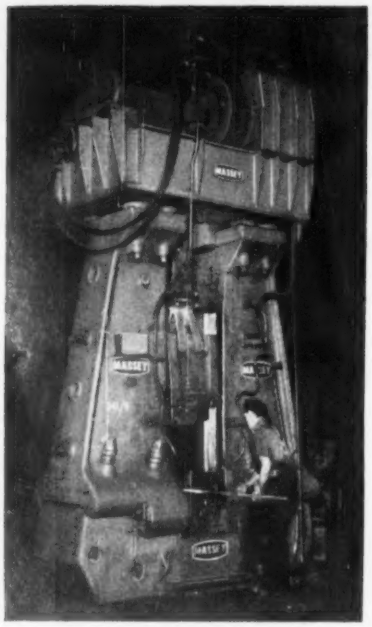

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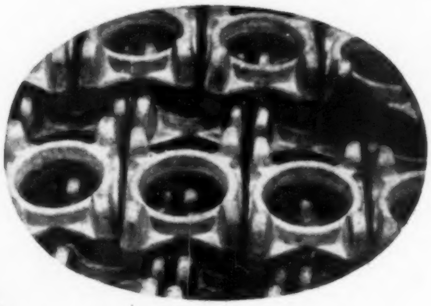

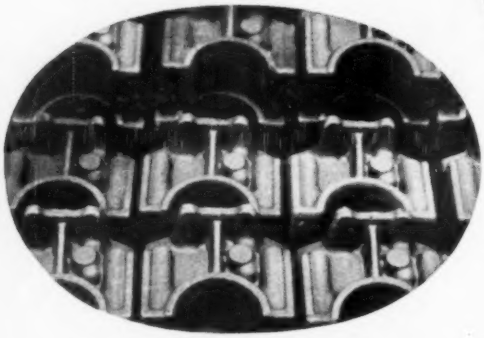

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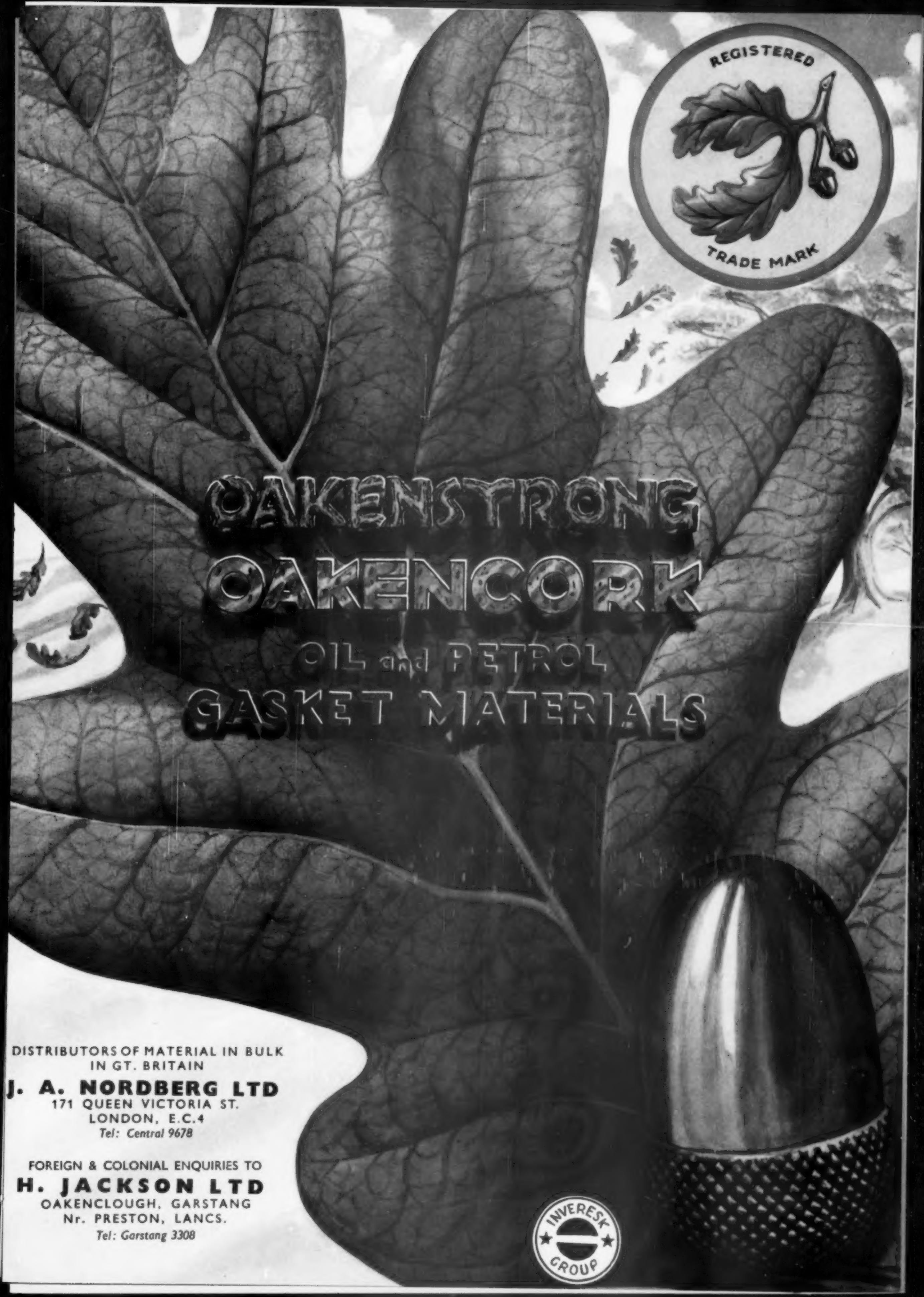
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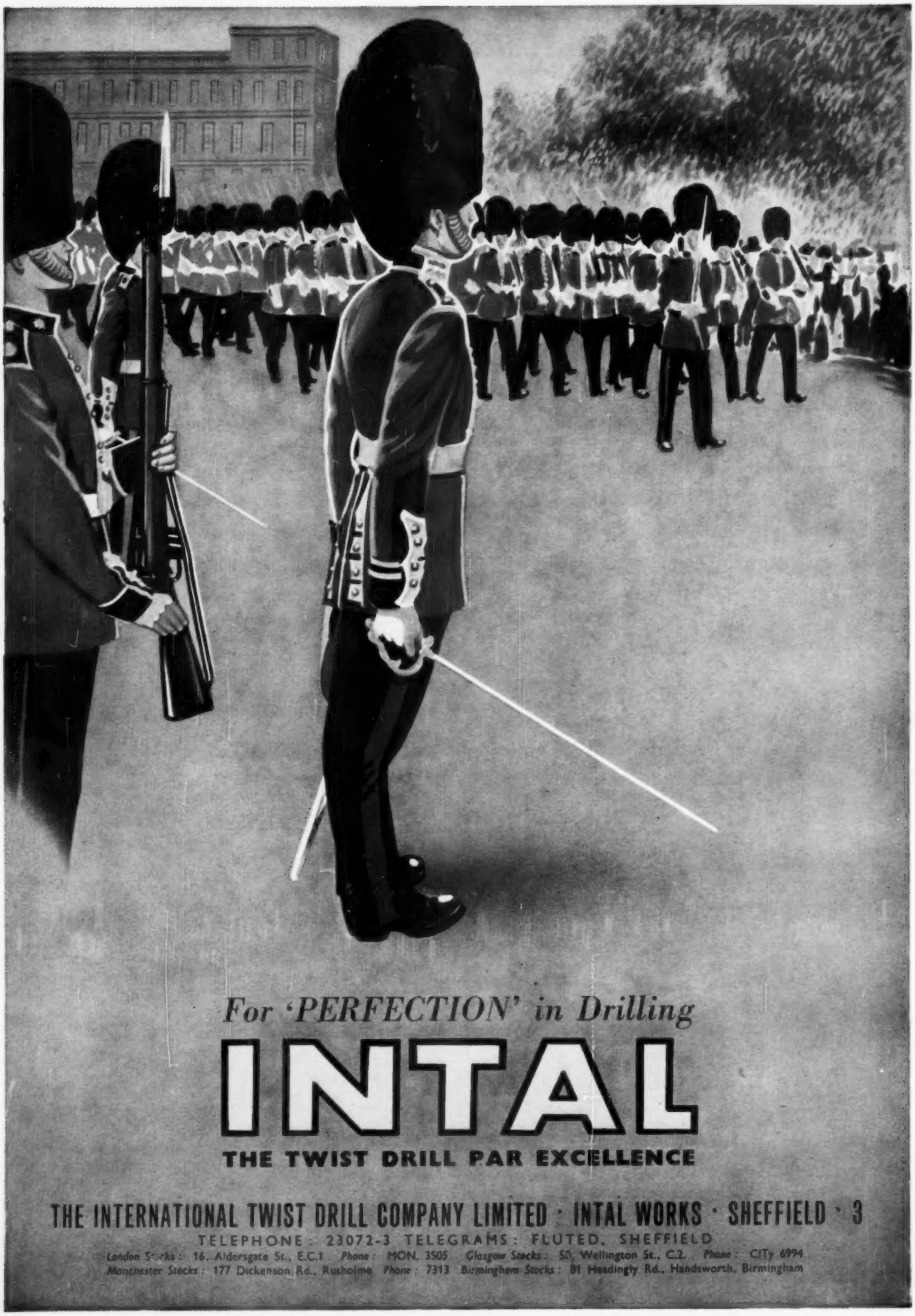
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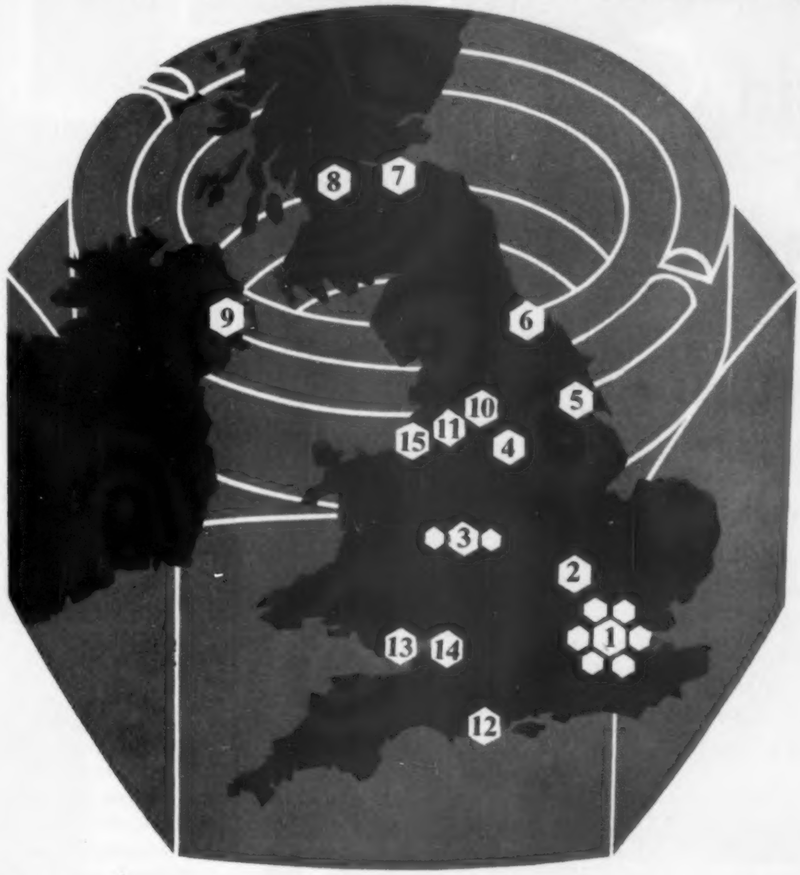

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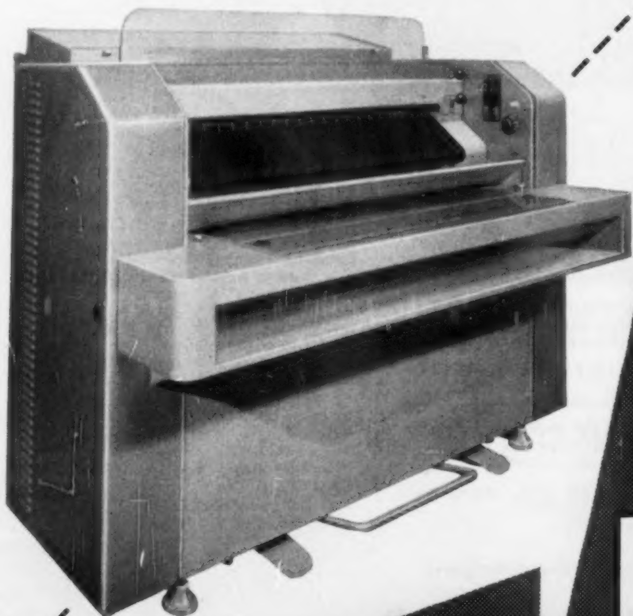

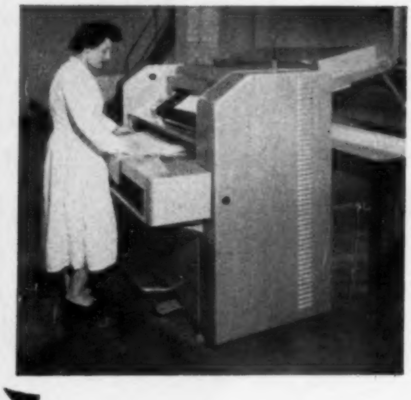

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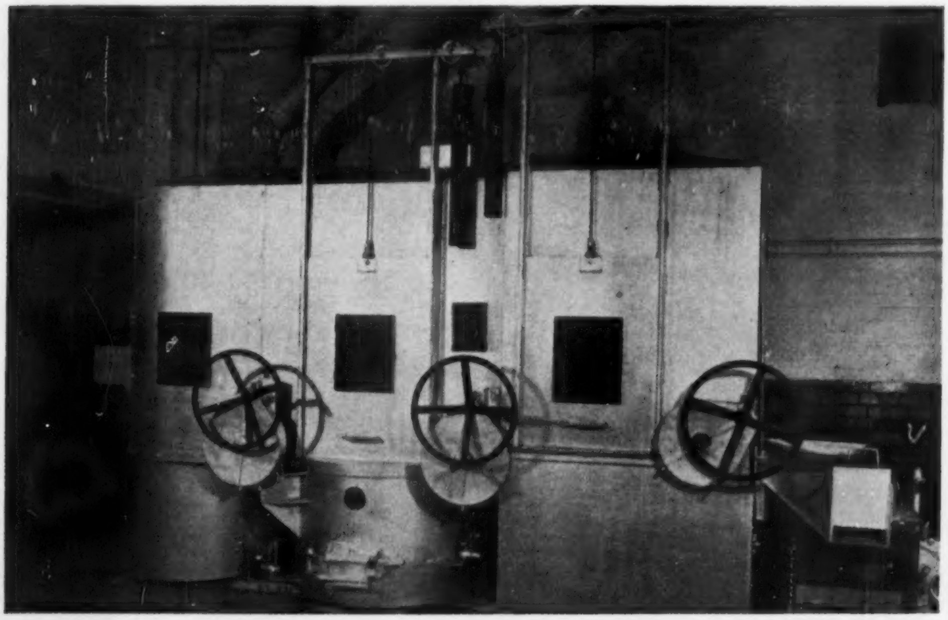

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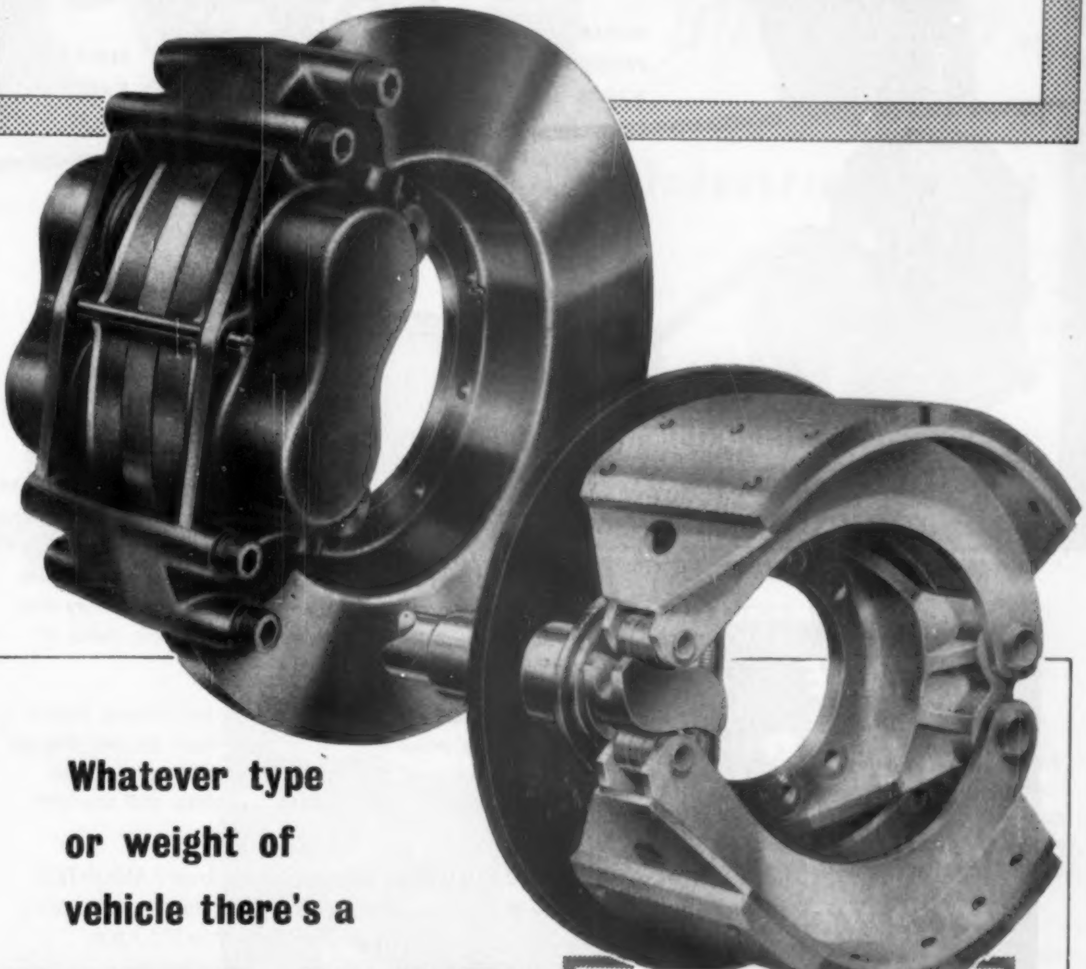

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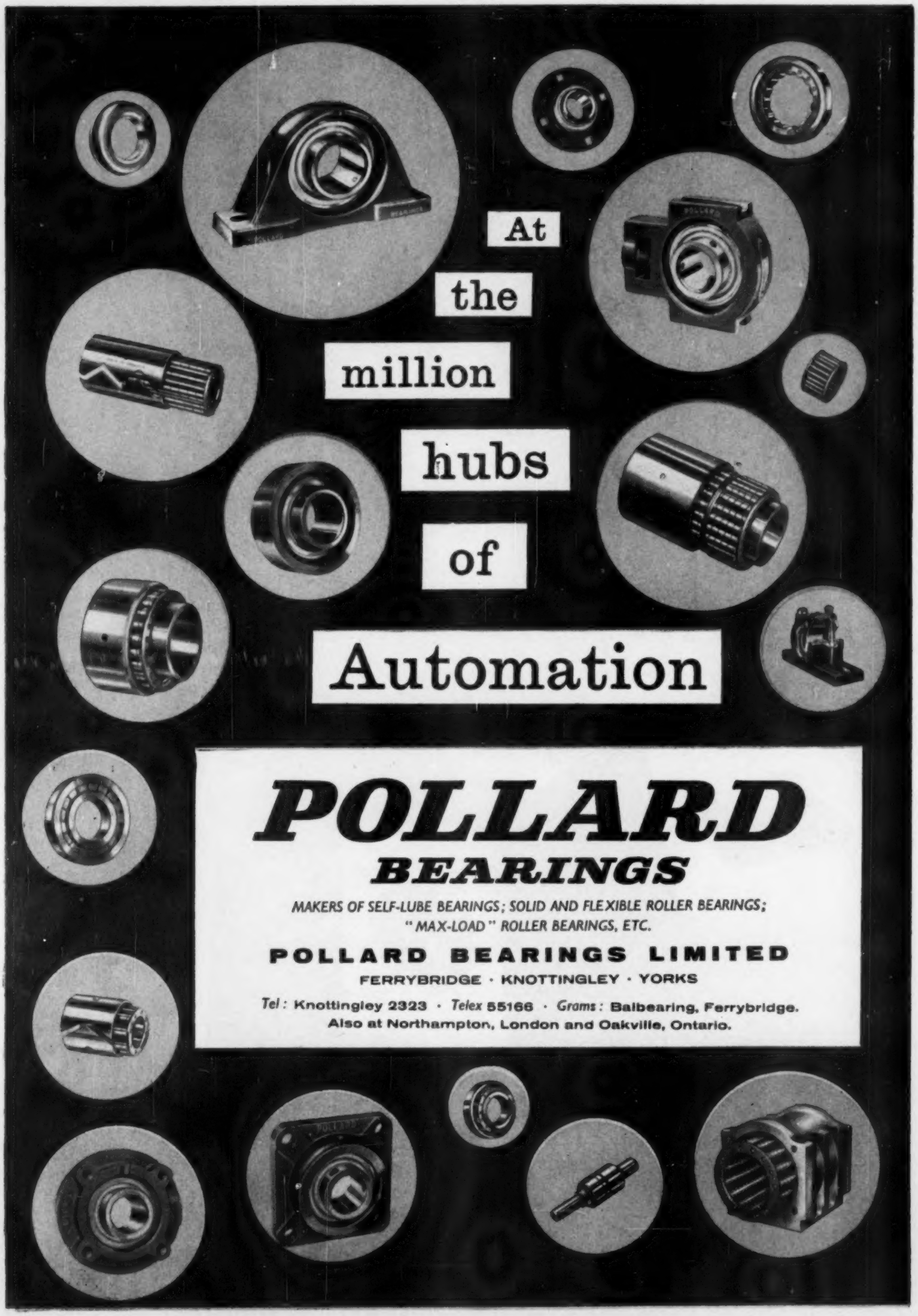
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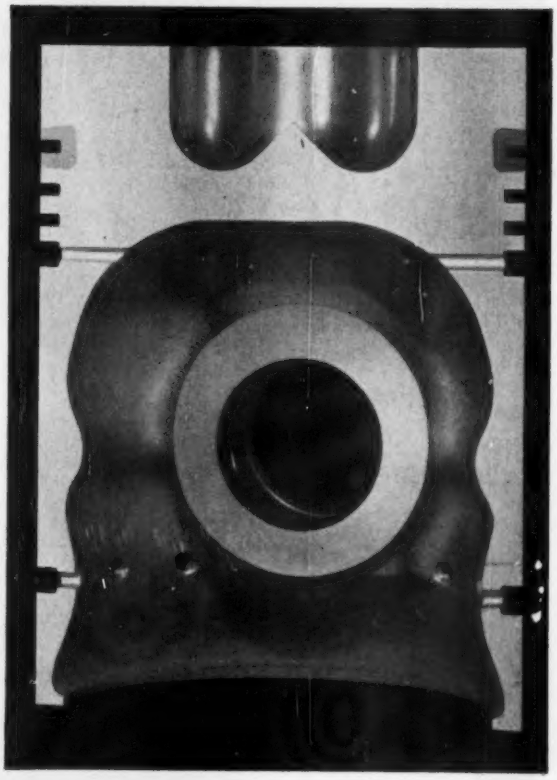

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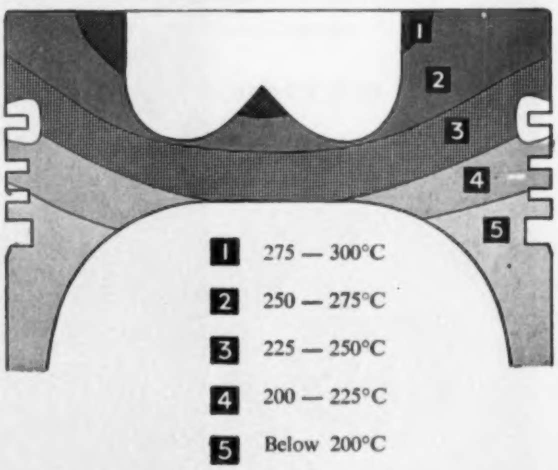

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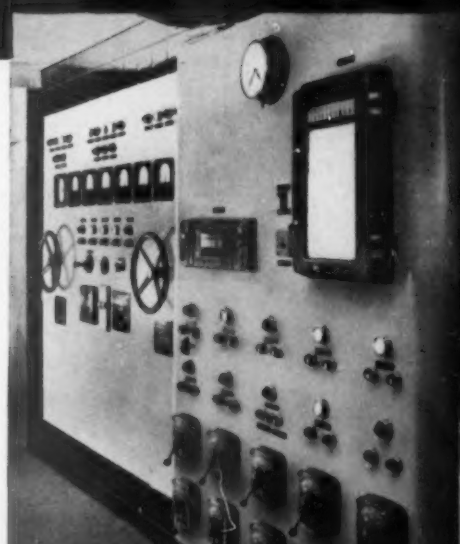

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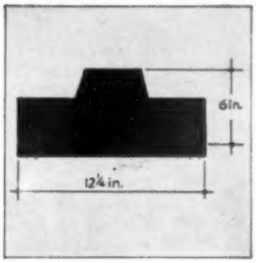

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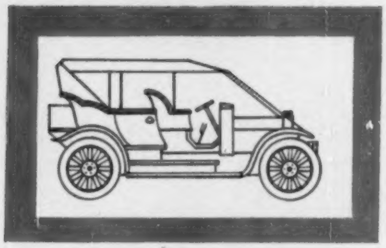
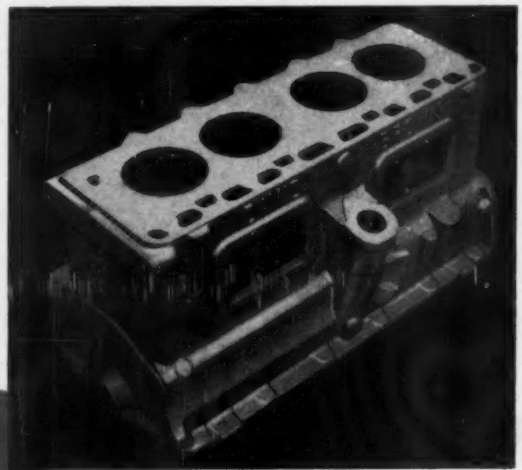


Cast iron cylinder block
for DB 25 tractor.
By courtesy of David Brown Ltd.



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AUTOMOBILE ENGINEER

CONTENTS

503 Editorial *Welfare Costs*

504 Bodywork Review *An analysis of the trends and improvements revealed by a study of the 1959 cars*

515 Mounting Anti-Friction Bearings *A study of the reduction of diametric clearance in bearings mounted with an interference fit*

519 Manifold Tuning E. M. Goodger and J. Hillsdon *Reassessment of the basic principles and potentialities*

523 Water-Cooled Brakes *An American system designed to reduce maintenance and increase braking efficiency*

524 Lateral Forces on Tyres D. H. Cooper *Distribution of side-force and side-slip in the contact patch of a pneumatic tyre*

528 Recent German Publications *Brief reviews of current technical books*

530 Simplified Sealing *Ingenious applications of Dowty synthetic rubber rings by the Ford Motor Co. Ltd.*

533 A New Ford Engine Plant *Full use of automation techniques for a production rate of 4,000 units per two-shift day*

536 Precision Barrelling *The development of new methods and processes by Rolls-Royce Ltd.*

541 Current Patents *A review of recent automobile specifications*

RINGS OF SYNTHETIC RUBBER CAN PROVIDE SIMPLE AND EFFECTIVE SEALS FOR STATIC OR MOVING JOINTS. HERE IS A DOWTY SQUARE-SECTION RING SEALING THE KING-PIN BUSH ON A THAMES TRADER VEHICLE

Published the second Wednesday in every month
by ILIFFE & SONS LIMITED
Dorset House, Stamford Street, London, S.E.1
Telephone · Waterloo 3333 *(60 lines)*
Telegrams · Sliderule, Sedist London
The annual subscription inland and overseas
is £3 0s 0d *including the special number*
Canada and U.S.A. $8.50

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Coventry · 8-10 *Corporation Street*
Telephone · Coventry 25210
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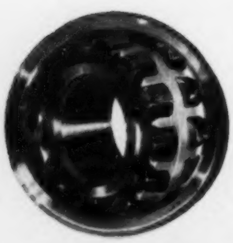

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DESIGN MATERIALS **AUTOMOBILE ENGINEER** PRODUCTION METHODS WORKS EQUIPMENT

Welfare Costs

WELFARE is today an integral part of the production process, but the value and the costs are difficult to assess. Services provided are extremely diverse in character, practices differ between industries, and between firms in each industry. One aspect has become common to all: the cost of welfare constitutes an appreciable addition to the annual wages and salaries bill.

It is desirable that a clear picture is presented of the costs of these amenities; of how costs are allocated; of their value in terms of production efficiency, labour relations, and the retention of labour; and of how the costs compare with those of other organizations at home and abroad. This last point gains topical importance by the imminence of the European Common Market and the possibility of the introduction of a European free trade area.

A comprehensive survey of the subject is currently being conducted by the International Labour Organization. In Britain, the report of a survey of the cost of welfare, entitled "The £ s. d. of Welfare in Industry," has recently been issued by the Industrial Welfare Society, Robert Hyde House, 48 Bryanston Square, London, W.1. A total of fifty-five firms, twenty-one in engineering, participated in the survey. They range from an engineering company with 250 employees to a chemical concern with 110,000. The period covered was the firm's financial year ending in 1957.

Factual information in the report is itemized, summarized, and presented in tabular form. Throughout, the data relating to the median firm is taken as being more representative than an averaged figure in a widely ranging survey of this character. The median firm is defined as that placed mid-way in a list of firms arranged in order of rank for the item under consideration.

In the overall summary the cost of personnel administration, as distinct from welfare, is shown as £11 10s. 4d. per employee per annum, or 1·98 per cent of the total remuneration, including cash bonuses and holiday pay. The cost of welfare benefits, however, is given as £84 5s. 6d., or 14·49 per cent of the total remuneration. The tangible welfare benefits are, to varying extent, contingent ones. They depend on whether the employee eats in the canteen, participates in social and sports clubs, is sick, or lives beyond the age of seventy. It is incorrect, therefore, to express the figures in a manner that suggests the employee receives almost £2 per week in benefits.

Expenditure on welfare in the engineering industry, 12 per cent, is lower than for industry in general and is contrasted with 18 per cent for the food industry. The size of the firm, in terms of the number of employees, cannot be correlated with the varying rate of expenditure. One fact emerges clearly—welfare benefits are not applied to supplement a low wage rate. As has been found in the U.S.A., the higher the wage rate the higher the scale of benefits.

In the provision of housing and health services Britain is ahead of European countries. As regards pensions, family allowances, sickness payments and holidays, many European countries are doing much more than is generally realized in Britain. To quote the survey—"West Germany is now, with its most comprehensive pension and sick pay scheme, a far more thorough welfare state than Great Britain". The rates of employer contributions to compulsory social security schemes in thirteen European countries in 1956, expressed as percentages of assessable wages, showed that only in Sweden and Ireland were they lower than in Britain. All the six European Common Market countries pay higher rates. From this it must not be inferred that British wage rates are exceptionally high; Switzerland and four Scandinavian countries had higher average hourly earnings.

These matters, and the question of how the cost of social security measures will be spread between the State, the employer, and the employee, will be of some importance in the years ahead, and particularly should unilateral operation of the six-country European Common Market develop. To attempt to divert the cost entirely to the State would inevitably result in a higher level of general taxation. This would mask the issue but must be regarded as inequitable, as it tends to bear unfairly on the efficient and prosperous firm.

The Industrial Welfare Society has done a notable service in conducting this survey and making the report available. It should be in the hands of all concerned in the organization of welfare activities and could be of great value to managements in providing reliable data on what is being done in this country and abroad. The motor industry is no laggard in these matters and, as this was being written, it was announced that a progressive company, employing more than 17,000 people in four factories, and building a fifth plant that will engage another 4,000, had launched a new non-contributory life insurance and pension scheme. This has the aim of "ensuring that the employees should always be better off than if they had to rely only on the minimum State benefits".

The new Austin A40 is well proportioned and has ample window area. By extending the roof almost to the rear, an impression of length is gained

Bodywork Review

An Analysis of the Trends and Improvements Revealed by a Study of the 1959 Cars

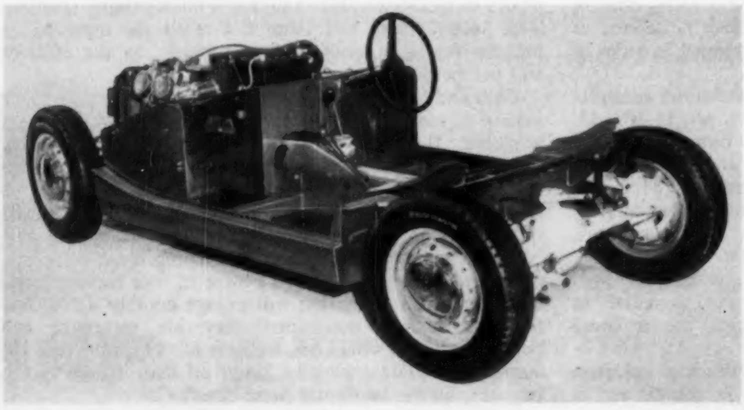
ONE of the most interesting trends in private car bodywork during recent years has been the divergence of design between the small and the large car. This divergence was even more pronounced than ever before at the 1958 London Motor Show. The design objective in respect of the smallest models is of course to make the best possible use of the space available, having regard for the capabilities of the engine. At the other end of the scale, the larger cars are constructed to provide the maximum of comfort for at least six adult people. Between these extremes is a wide selection of family saloons, drop-head coupés and sports cars; bodywork is available to suit all tastes while the size, appointments and the quantity produced form the basis on which the price is determined.

A study of the latest designs reveals that engineering technique is by no means confined to the mechanical parts of the latest cars. Rather is there evidence of an increase in the number of chassisless models designed on sound basic principles. In this form of construction, it is important to spread the loads, rather than to take them locally at the different application points. Also the design must be such that the stressed panels are stabilized against bending or buckling. Torsional stresses are taken by the inside and outside panels, while the roof pressing and the door pillars all take their share of the overall bending loads, which are also spread over the door sills, the propeller shaft tunnel and various stiffeners.

Variations on this basic theme include the mounting of the engine and front suspension on a separate sub-frame under the front of the body shell. Alternatively, the body sides and other panels can play their part in the overall stiffening of the body structure and can carry the engine and front suspension; this means, in effect, that only the bonnet, boot lid, and in some instances the doors, are unstressed parts. The constructional method chosen may have some bearing on the ultimate outward appearance of the car.

In contrast to British and Continental practice, most of the American cars have a separate chassis, so that the bodywork takes little or none of the main loading. This fact has a direct bearing on the facility with which frequent changes of bodywork styling can be made.

A year ago, only three British cars featured wrap-round windscreens, but, with the increasing availability of this type of windscreen glass during the interim, the number of cars so fitted has greatly increased. This change has brought about an increase in the number of cranked screen pillars, with



This view of the basic structure of the Austin-Healey Sprite shows how the engine compartment extends between the leg wells

consequent alteration of the fundamental design of front doors. From the rapidity with which the series production bodywork has so far changed in this respect, it is reasonable to assume that the trend will be extended to the majority of new bodies now being designed.

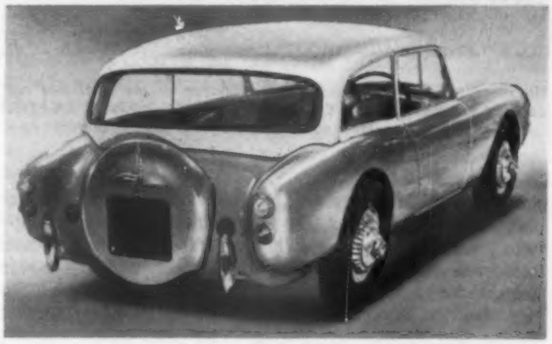
One of the latest examples in the unitary construction sphere is the Humber Super Snipe. The all-metal body, which is basically the same as that of the Humber Hawk, is so designed that it provides great strength with a low centre of gravity. Interior accommodation is provided for six grown people, and ease of entry has received particular attention. With a wrap-round screen and cranked pillars, there is inevitably some reduction in front door opening width above the waist-line. However, in this instance the wheelbase of the Humber Super Snipe is sufficiently long to avoid excessive encroachment on the width of the opening. This indicates that, provided that there is sufficient body length, this latest design of screen can be used without obstructing entry, but that if it is to be employed on cars of shorter wheelbase, then there is a strong case for the employment of only two doors.

The introduction of the cranked windscreen pillar has brought with it the problem of water running down the outside of the pillar above the waist-line, seeping through the gap between the door and the pillar, and ultimately dripping on to the floor inside the car. Disposal of such water has been catered for in most designs by the introduction of a separate water channel, leading forward and downward from the rear of the horizontal portion of the pillar; the channel runs just below the waist-line of the scuttle and disappears behind the front wing pressing. Normally, the rear end of this water channel is concealed behind the closed door. Thus, when water runs from the roof down the pillars and the outer edges of the windscreen, and seeps through the door break-line, it is caught by this channel and led away to the ground. An entirely different method for the disposal of water from the upper part of the car is employed on the Panhard Dyna de luxe saloon. Here, the roof water channel extends along the sides of the roof above the doors and then drops downward at the pillars to a point well below the waist-line of the car.

One of the few new examples of the chassisless method of construction is the light-weight integral structure of the Austin-Healey Sprite. Underneath the body panels and well forward in the wheelbase is a pressed steel scuttle unit, taking the form of two separate leg compartments and toeboards, one on each side of the engine installation; this structural unit extends back as far as the instrument panel. At the top of the scuttle are the fixings for the windscreen frame and the bonnet hinges, and at the bottom, the assembly is attached to the main structure comprising the two body sills, the propeller shaft tunnel and the main floor pressings. Two L-shape, top hat sections beneath the seats, extending from the central cross-members to the rear portion of the structure, provide part of the support for the luggage boot, in which are carried the spare wheel and the petrol tank. Additional support for the rear end is provided by the stressed panel over the boot.

The bonnet, front wings and grille form a single fabricated assembly that is rather a heavy structure to hinge on the scuttle. When it is in the down position, a forward cross-member supports the hinged portion. The doors of the Sprite are hinged at the front and secured to the shut-pillar by interior locks, devoid of outside operating handles. To open the door, it is therefore necessary to release the catch inside the body; this means that, when the hood and side-screens are in position, the hand has to be passed through the side-screen flap. Since the body is an open two-seater, there is little point in using the lockable type of catch with an outside handle; the inside handles can be reached in any case.

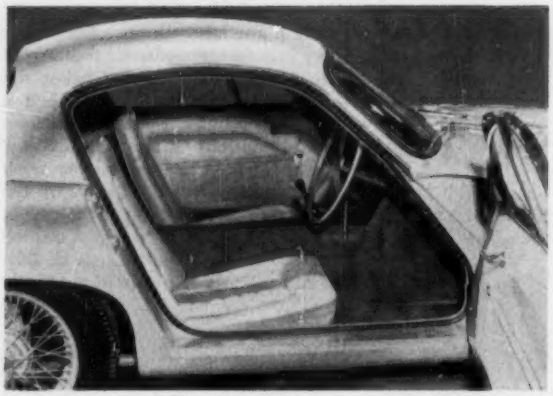
Popular in pre-war cars, the sliding roof has completely disappeared from production models. In its place, as a means of ventilation, many cars feature some form of styled grille, which takes up much of the scuttle panel below the windscreen and forms the intake for fresh air. Such a location has the advantage of avoiding the entry of exhaust fumes from traffic ahead. An example of this type of intake is found in the new Rover 3 Litre saloon, where the fresh air is channelled into the body through two grilles set in the edges of the roll above the instrument panel, at the corner curves of the windscreen. The fresh air supply can be controlled by a neat lever close to the grille and can be shut off when it is not needed; it is quite independent of the heater and demister equipment. To complete the interior circulation, air extraction is assured when the windows are lowered: above each is a louvre, to avoid draughts or the entry of rain.



Constructed of reinforced plastics, the latest Berkeley four-seater is available either in open form or with a hard top. As the lower view shows, accessibility with the bonnet raised is excellent

A new method of construction for metal bodywork has now been introduced by Aston Martin Lagonda Ltd., in the form of the new Aston Martin DB4. The aerodynamic bodywork is built up in two major assemblies. Below the waist-line is a series of sheet steel pressings comprising the floor between the front bulkhead and the luggage boot, a sturdy front bulkhead, a pair of rear wheel arches and a rear bulkhead; these pressings are reinforced where necessary for rigidity. Under the wide door openings are sturdy box section sills.

Above this assembly is a light-weight structure of small



On the Lotus Elite two-seater, the inclination of the front and rear edges of the doors is such as to provide relatively easy entry and egress for the occupants, in spite of the very low build of the car

diameter steel tubes, welded to form the framing of the upper part of the bodywork. At the front is the frame for the wrap-round windscreen, from each side of which the cant rails sweep rearwards. From this major framing extend the rear window surround and the rear wing tail-piece framings; the latter are bridged across the bottom of the boot by a further light-weight tubular framework. The front pillars support the large windscreen, but the doors are hinged to the main chassis structure, and their weight is distributed to the front bulkhead, to which the A-pillar is secured. This system of chassis and body frame construction was evolved some years ago by Carrozzeria Touring, of Milan, and has been thoroughly tested. The Aston Martin body is panelled in light alloy pressings of very attractive form.

The new Austin Hirecar consists of an all-metal body built by Carbodies Ltd., on an Austin taxicab chassis. This body has four doors and, as well as providing a high standard of comfort for the passengers, introduces a number of interesting features and materials. Since it is a hire-car, there is a division between the driver's and passengers' compartments, and the top of this division comprises two fixed glass panels. Let into the panel on the driver's side, at a point near his ear, is a circular aperture made of white plastics material, which is fitted with a rotary shutter. The assembly is secured to the glass by three bolts and the shutter can be opened or closed by either the passengers or the driver.

In the rear compartment, on each side of the division and just inside the doors, are two sturdy half-pillars, which prevent passengers from bumping their shins against the folded occasional seats, as they enter the car. These occasional seats are bolted to the floor and, in the stowed position, are secured to the division by catches. The rear seat can accommodate three passengers, and on the model on show, two additional passengers could be seated on the folding units. According to information gathered at the recent London Show, however, these separate seats may be replaced by a single bench type seat.

The door trim throughout is of the new moulded Royalite material, which is smooth and very easy to clean—a point of particular importance to hire-car operators. For the seat trimming, high-quality hand-buffed leather is employed over foam rubber foundations, and the rear seat is built up on a board base. The heater unit is in the rear seat pan and is well set back to prevent its being damaged by passengers' heels. A noteworthy feature is that it provides warm air at foot level. A switch on the off-side centre pillar controls the heater and can be operated by the passengers.

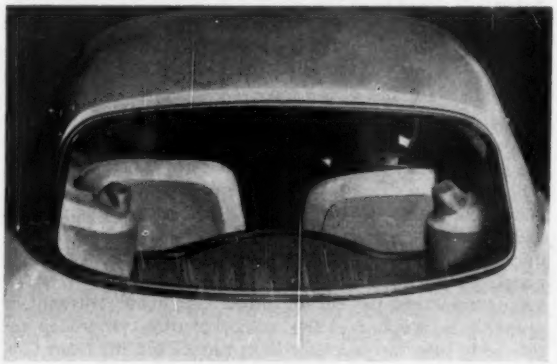
In the driver's compartment, a seat is provided on the near-side to accommodate an additional passenger and, when not in use, it can be folded up to the squab to provide an unobstructed floor space for luggage. A large boot behind the rear seat augments the luggage-carrying capacity; the boot lid is hinged at the bottom and is supported by hinged stays when open. The spare wheel is stowed flat against the squab board of the rear seat and is secured by a large wing nut. Only a few turns of the nut are needed to release the wheel, because the retaining bracket is designed to slip through the central hole in the wheel. The driver's seat is adjustable, both back and forward and for height, over a wide range. It has a foam rubber cushion and squab, covered with hide.

At the front of the vehicle, the grille is integral with the bonnet, and the bonnet pressing is braced by steel tubes welded in position underneath the main panel. The bonnet catch is set low down and is formed by an external sliding trigger between the bumper bar and the radiator grille. All the doors have drop-windows, which can be opened and shut by pressure on a length of metal section secured to the top of the glass: the mechanism is balanced so that the window has no tendency to move from its set position. The absence of handles or catches makes for a minimum of projections on the inside of the doors, and so reduces the risk of injury to passengers.

Plastics bodywork

Body designers are not confined to metals in their choice of structural materials and considerable progress has been made in the use of glass fibre reinforced plastics materials in the form of mouldings. One company with much experience in this field is Berkeley Cars Ltd. Their latest model is a four-seater body which, like their sports two-seater, is available with either a soft or a hard top. All the Berkeley cars are constructed on the chassisless principle, the body and frame being of polyester-glass plastics, into which bulkheads and cross-members of aluminium alloy are moulded during the laying-up stage.

Another feature of the Lotus Elite is a large parcels shelf behind the seats. The protruding tops of the suspension units are trimmed



The bonnet of the new four-seater model is a one-piece moulding, hinged at the scuttle and incorporating the frontal grille. This design gives complete accessibility for servicing and overhaul of the engine, front suspension and steering. The bonnet is secured at the forward end by a pair of spring-loaded trigger catches operating in a metal tube moulded into the bonnet; it can be held open by a light-weight rod fitting into a clip on one side of the body. At the rear, the spare wheel is mounted on the outside of the body and covered with a polyester-glass moulding. To this cover is fitted the rear number plate, together with its

illumination lamp. The wiring to this lamp is fitted with a two-pin plug, which can quickly be disconnected when the cover is removed.

There are no exterior door handles on Berkeley cars, and an interesting lock arrangement is used on the two-seater hard-top body. The side windows are made up and bound in a plastics covering material and glasses are inserted, the rear glass being fitted inside the front glass and left free to slide. The doors can be locked from the inside when the driver gets out of the car. To unlock the doors, the sliding section of the glass is pushed forward until there is sufficient space for the hand to be inserted and the key fitted into the lock.

A new small car, as yet untried under service conditions, is the Stirling, manufactured by S. E. Opperman Ltd. This is another polyester-glass plastics structure of clever design, without a separate chassis. It mainly comprises two major mouldings, with a horizontal joint, and two door mouldings. The body is designed to carry two adults and two children. The lower half of the body is a large boat type moulding, in which steel reinforcements are embodied during lay-up; these reinforcements carry the main loads of the engine, transmission and wheel suspension. There are no orthodox bulkheads, but a sturdy parcels rack across the forward end of the lower moulding helps to take the front suspension loads. All the window glass is in the upper moulding, which comprises the roof dome, instrument panel, bonnet top, boot lid and the upper portions of the front and rear wings. The joint line of the two main mouldings is about half-way up the body, and each moulding incorporates an outward-turned flange to make the joint. At the rear of the car, the joint occurs at the level where normally a boot lid would join the body, while at the front end it is below the head-lamps, at a level where generally a lifting bonnet joins the grille.

Two large mouldings form the doors, which give good access to the front seats. When the rear seats are not occupied the squab can be folded down to provide additional luggage space. The layout stems from the rear installation of the engine and transmission, and indicates great possibilities for the future of polyester-glass plastics to car construction.

Jensen Motors Ltd. were the first to manufacture a high-performance saloon car body of polyester-glass plastics, and basically the body design of their 541 model has remained unchanged in the 541R. An interesting detail of this car is the inverted air extractor grille, which is just below the top of the bonnet. This grille has been incorporated in the bonnet so that it does not interfere with the aerodynamic shape of the car. It improves cooling efficiency by avoiding any build-up of air pressure inside the engine compartment.

Another application of polyester-glass bodywork, to a light chassis, is the Frisky small car, made in convertible and hard-top forms. Several changes have recently been made to the styling of these models, to facilitate the moulding of the body-work; among them is the use of a much simplified, one-piece moulding for the rear end. The new Friskysprint is designed to conform to international competition regulations. Its bodywork comprises a series of mouldings, bonded together, and embodies deep sills at the base; two small drop-flaps are incorporated to facilitate entry into the vehicle.

A newcomer to the Show was the plastics-bodied Peerless G.T. coupé. This body is, in fact, built up of 57 separate polyester-glass mouldings bonded together into a complete body unit. The boot lid is of unorthodox design and takes the form of a large flap in the lower tail of the body rather than a lid. Pivoted on two lift-away hinges at the top, the flap lifts upwards to give access to the boot or the spare wheel.

Another car with integral body-chassis construction, of reinforced plastics, is the Lotus Elite. This two-seater, two-door body has a large wrap-round windscreen and rear window, and the door windows have hinged quarter panels for ventilation. Behind the two seats is a large parcels shelf, on each side of which the upper ends of the rear suspension units protrude into the bodywork, where they are readily accessible; the protruding portions are trimmed and topped with a small cap. Space for luggage is provided in the rear of the body.

Shallow roof pressings

Another outstanding design feature that is being rapidly adopted on most saloon models relates to roof pressings which are becoming progressively shallower. This feature is very noticeable on the new Austin A40, on which the pressing has a slight transverse convexity over the central portion, terminating at each side in a moderately sharp radius and then a short drop to the cant rail. In addition, the roof extends slightly aft of the wide and deep rear window. The use of shallower roof pressings means that the glass height must be increased since it is undesirable to reduce the interior space between the floor and headlining to less than about 48 in.

Not only has the new Studebaker Lark a shallow roof pressing, but also it is provided with one long water channel along both sides of the roof and across the front above the curved windscreen head-rail. This means that normally, water does not run off the roof and down the windscreen, but is channelled to the rear of the car, above the wrap-round rear window, and dispersed by the airstream.

Fold-down doors are an unusual detail of the Friskysprint two-seater competition car

Below: On the Peerless G.T. coupé, access to the boot is through a vertical opening

A study of this body discloses a new relationship between the doors and their pillars. Normally, the width of the upper parts of the pillars is related to the door opening line and is the same at the top and bottom. On the Lark, however, the centre pillar, which is visible between the front and rear doors, tapers from waist to roof. As a result, the rear edge of the front door glass is inclined rearwards, while the front edge of the rear door glass is, in fact, vertical, as viewed in side elevation. This produces two break-lines, which remain parallel from the bottom of the doors to the waist and then converge until they almost meet at the roof-line.

This altered relationship appears also at the rear of the rear door. The quarter pillar that forms the upper portion of the door jamb leans back from the waist, but the break-line is vertical, so that in fact it cuts across the pillar from top to waist. The door edge continues downwards on the same line and, when level with the top of the rear wheel opening, encroaches slightly into the rear wheel arch, so that there is only a small radiused area, or rocker panel, above the door sill. Surprisingly, the whole styling of this car is European in character rather than American: the front overhang is limited, while the rear end is completely devoid of fins and massed rear lamps.

Another approach to the styling of the rear portion of the roof is on the Lancia Flaminia. Here, the rear quarters are stiffened by a deep fin extending from the top of the roof to the rear of the boot; it is almost an L-shape as viewed from the side of the car. The very deep rear window has the appearance of being sunk between the fins and is fitted with a pair of windscreen wipers. Such radical departures from the traditional design of roof pressings, while playing their part in the overall styling of the car, are also desirable to stiffen shallow roofs of large area.

Body side panels

With slab-sided bodywork, the actual position of the waist-line is in many instances rather indeterminate, especially when it blends with the crown line of the wings. Nevertheless, the slab-sided style appears to be holding its position. Examples of this type of bodywork include most of the Ford range, the new Rover 3 Litre, the Riley and the Wolseley models. On a number of cars, the rather plain appearance has been counteracted by introducing shape into the panels somewhere along the length of the car, thereby providing well defined boundary lines.

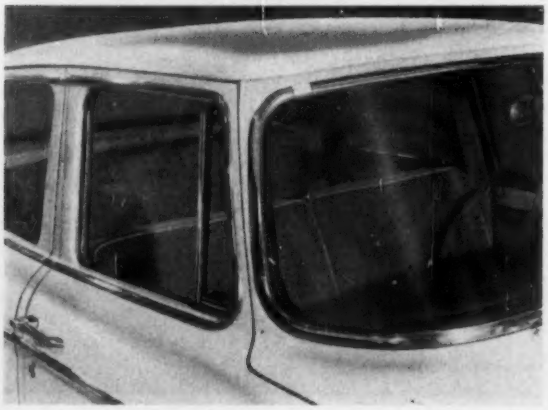
Examples of pressings with pronounced relief are the side panels of the Standard Vignale Vanguard. A pressed line, forming the boundary of a depression, rises up and forward on the front door, and is then swept back and continued over the rear door, and above the rear wheel opening, finally merging with the main surface at the rear of the car. In addition, the edges of the wheel openings are swaged outwards to provide the stiffness required round this unsupported section. A different, and less pronounced, relief effect can be seen on the Vauxhall Victor, in which a sweep in the top portion of the rear door joins what is almost a V-section on the rear wing.

Distinct lines that conform to the more orthodox wing profile are retained on the new Jaguar Mark IX; the line sweeps downwards from the crown of the front wing to the lower part of the rear wing, the contour of which it then follows. The same principle is applied to styling of the new Armstrong Siddeley Star Sapphire, and also to the Daimler Majestic, but to a lesser degree, in that the front wing line merges into the main shape at a point some distance along the front door. The slab-sides of the Humber Super Snipe are relieved by a step in the waist-line at a point half-way along the rear door, and the wheel openings are swaged outwards to provide stiffness.

On the Sunbeam Rapier, there are slightly out-turned tail fins, extending from the rear door to the back of the car, to enliven the appearance. In addition, on many models, rubbing strips and mouldings are employed in a number of ways to provide break-lines. A feature of the Vauxhall Velox and Cresta models is a wide moulding that starts at the headlamp hoods and continues along the complete length of the bodywork. In contrast, the Austin A55 has an applied side moulding that dips sharply in the centre of the rear door and then rises and curves away to the rear of the car. Moreover, this model has a raised rubbing strip over both wheel openings and extending the full length of the body sill.

Rear-end treatment of the Studebaker Lark is restrained. The opening lines of the doors are obliquely disposed on the pillars

A noticeable current tendency is towards shallower roof pressings. An example of this styling is found on the new Studebaker Lark



Bumpers

Although the standardization of the height of bumper bars remains unsettled, the actual layout of these bars on British cars appears to have stabilized in the designs that just turn round the sides of the bodywork. Practically all models now fit this style of bumper, with or without over-riders. The possible future trend in respect of bumpers is indicated on the new Austin A40, in which the rear bumper extends for a distance along the sides of the body and conforms closely to the body shape. This bumper has flat portions and holes for the mounting of the number plate and its illuminating lamp. On the Austin-Healey Sprite, there is no rear bumper, but a pair of large over-riders is secured to the back of the body—an arrangement that has previously appeared on other sports cars in both this and other countries.

Right: The Humber Super Snipe has the same body shell as the Hawk and has good proportions

Below: On the Lancia Flaminia, fins on the rear wings are taken upward to the roof line

The over-riders of the Simca P60 Aronde embrace large oval rubber buffers, which protrude about 1½ in ahead of their casings and some 4½ in from the bumper bar. One of the advantages of using rubber at this point is that when these buffers come into contact with another vehicle, there is no chromium plating to be damaged at the point of impact. In contrast, several of the 1959 American cars have their fog lamps in the bumpers, near to the ground and arranged to direct their beams either to the centre or to the side of the road.

Conceptions with regard to frontal styling and the shape of radiator grilles vary widely. Although a number of cars—such as the Ford models, the Hillman Minx and the Morris Minor 1000—have the wide and shallow style of grille, the vertical styling is retained on the Jaguar, Armstrong Siddeley, the Wolseley and Riley ranges, and on the Austin A35. In addition, there are the almost square grilles on cars such as the new Austin A40, the MGA coupé and the Bristol 406. The slit type of frontal styling, that started with the Citroën D.S.19, is found also on the Lotus Elite coupé and the Ferrari 250 Granturismo coupé, on which it is very pronounced.

The spare wheel

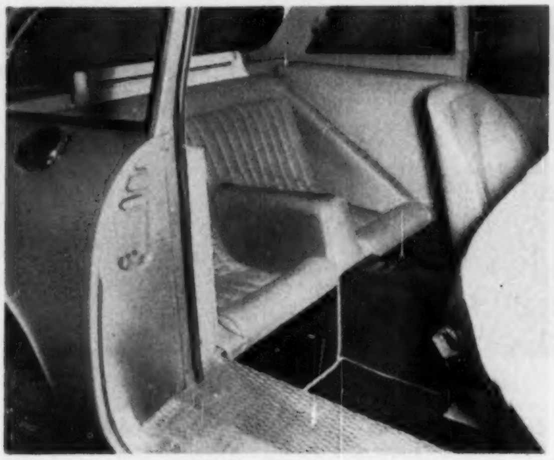
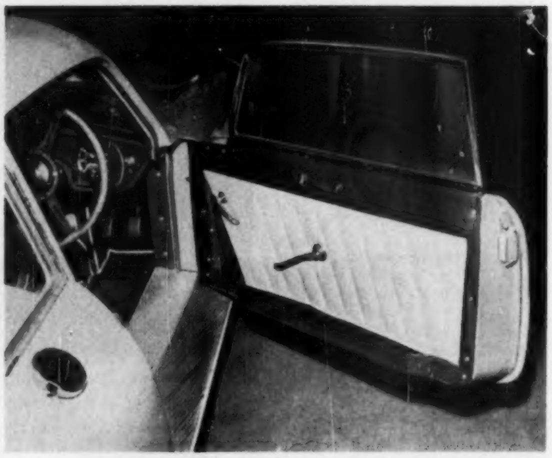
Stowage of the spare wheel is always a problem, particularly since accessibility is so important. On many American cars, despite their size, there is a shortage of boot space, because the spare wheel is merely placed on the floor of the boot and secured there. If the wheel is to be inside the boot, then it should be accommodated vertically. One ingenious arrangement is that of the Simca Vedette, in which the wheel is housed in the hollow of one rear wing fin. Another approach, used on the Bristol 406, is to stow the wheel vertically within the normally unused space between the scuttle and the front wheel. Access is provided by constructing the rear portion of the near-side wing in the form of a large flap, which is hinged at the top and lifted up when the wheel is required.

Provided that there is enough room, a practical solution is to carry the spare wheel horizontally under the floor of the boot, where it can be secured by an angle-iron carrier. The carrier is generally hinged at its forward end and is lowered or raised by means of a long bolt, used as a screw jack, fitted into the floor of the boot. Such a system is employed on the Vauxhall Velox and Cresta models; the operating bolt head is close to the rear of the boot and is turned by the wheel-nut spanner. Provided there is enough room to operate the spanner, it is not necessary to unload the luggage from the boot in order to free the spare wheel. In the ample boot of the Armstrong Siddeley Star Sapphire, the spare wheel is in a separate compartment beneath the boot floor, an arrangement that permits the spare wheel to be removed without disturbing any luggage.

Colour schemes

An interesting analysis of the colour situation at the London Show was carried out by the Paints Division of Imperial Chemical Industries Ltd. It revealed a still further decline in the use of black as an exterior finish, with a marked increase in the use of shades in the ivory and coffee range. The popularity of red and maroon, although still high, had fallen by nearly a fifth and the greens had suffered a considerably heavier eclipse, partially compensated by an increased use of blue shades. Used on their own, metallic

What appears to be a spare wheel cover on the Plymouth Fury is, in fact, a swaging to stiffen the unsupported area of the boot lid

These two views of the Peerless G.T. coupé illustrate the deep floor wells and the width of the door sills, which contain the petrol tanks. The rear seat is a one-piece, hammock type moulding in reinforced plastics. Some elbow room is given by inclining the door trim panel below the waist-line

finishes appeared on a larger number of cars than last year but this finish found less favour as part of a two-tone scheme.

There is no doubt that, as a whole, car manufacturers are giving increased attention to colour finishes as a sales feature. Two-tone and even the so-called three-tone exterior finishes are now common and, rightly, the colours are chosen to enhance the overall styling and to emphasize certain features thereof, rather than to disguise faulty or misplaced lines. Where single colours are used, the appeal of the more vivid hues appears to be increasing.

Three-tone schemes are offered on the new Standard Vignale Vanguard; one such scheme features a coffee colour roof, off-white for the mid-portion, including the bonnet and boot tops, and coffee colour again for the lower portion. An alternative choice comprises coffee and pink. The range

Polished wood is used to a marked extent in the interior of the Jaguar Mark IX. Between the tables is a lockable compartment

of two-tone colours for the new Humber Super Snipe includes two shades of green, black and blue, and two-tone grey; several single colour schemes, including metallic blue and grey, are also available.

Typical of the lower-priced family saloons, the Hillman Minx de luxe can be obtained in grey and red, grey and blue, grey and green, powder-blue and charcoal, and other two-tone finishes with exotic but non-descriptive names. The same colours are available for single-tone finishes. Two of the new range of colours available for the 1959 Vauxhall cars are ivory and a special shade of pink known as Royal Glow; another is ivory with silver-grey.

Body design features

If the body of a car is to be truly functional, certain basic principles must be observed by the designer. Among these principles are: the body shall carry two or more persons with a certain amount of luggage, and that, at the same time, the body must permit reasonable access to mechanical parts for overhaul and servicing. On that basis, there are several current designs, both among the cheaper and the more expensive cars, that cannot be regarded as entirely successful.

One of the most difficult of all features of the design of a saloon is to obtain a low overall height without restricting the room inside the car. This problem is particularly intractable in the sports coupé style of bodywork. To enter a very low hard-top car is always something of an acrobatic feat, and such entry should not be further impeded by the need to climb over a very deep door sill. Deep foot wells are needed in many instances to obtain a low roof-line, and these involve the employment of a deep propeller shaft tunnel in both the front and rear compartments of private car bodies.

In some of the new cars, access to the luggage space in the tail is obtained by hinging the seat squabs. However, with this arrangement, it is essential that the opening into the boot is not appreciably obstructed by the rise in the floor necessary to clear the rear axle casing; otherwise this feature may make it difficult to pass cases into even the most adequate boot space. The danger of baggage being damaged as it is slid over such obstructions should also be considered.

A feature of bodywork design that is particularly undesirable for wet weather travel is excessive sweep of the sides into the tail, viewed in plan. On some well-known cars, this sweep is so sharp that much of the rear periphery of the tyres is not adequately covered by the wing line. With such a design, water is bound to be flung over following vehicles

and will also be turned back, in the turbulent wake, on to the rear of the car.

The ventilation panels on front door windows, in many cases, could be improved. Generally, these panels comprise a piece of glass in a frame pivoted on the door and surrounded by a rubber sealing strip. When they are open, a considerable draught may be directed on to the face of one of the occupants; moreover, in a high wind they may allow rain to be driven into the car. In heavy rain, some are by no means proof against the entry of water, even when closed, especially after lengthy service.

Entry to the rear seats of the Volvo 122S is made easy by the shallow door sills and the forward location of the rear seat

Finally, the catches on most of the standard products, in the form of a small cam lever, which is pivoted to wedge against a vertical post, are of little use against a burglar. It is a simple matter to insert a thin blade and lift this catch upwards, thus allowing the arm to be inserted and the door unlocked. On the Volvo 122S there is a cam fastener for these panels that cannot be turned until a central button is depressed from inside the car.

An appraisal of the 1959 American car models, to the extent of analysing the variety of shapes and decoration, indicates a number of trends that could be expected to become part of the basic styling in due course. As mentioned earlier, relatively frequent major changes in the body styling of these cars is possible, largely because the body is mounted on a separate chassis and therefore does not have to be designed as a stressed structure. With a stressed body, modification of styling features must frequently be confined to points where the basic strength of the structure is not affected. For example, changes could be made to the bonnet lid by the use of a double-swaged line in the place of the present single and centrally positioned swage. Properly carried out, this feature would stiffen this area of metal considerably. It is expected that paired headlamps will be adopted by British manufacturers in the relatively near future, and these undoubtedly will result in marked changes in frontal styling.

The Armstrong Siddeley 346 is one of the few limousines in series production. There is ample leg room below the occasional seats

The upper halves of the rear wheels on some American models are concealed by a long, rectangular wheel cover, so as to avoid a break in the lower skirt line without permanently boxing in the rear wheel. There is also a tendency to retain the very wide rear window, but to reverse the rake of its sides by taking it further up into the roof dome, while keeping the line across the body relatively straight and bringing the rear seat squab rather further aft. Some new British and Continental designs show a trend towards the elimination of the ventilation panels on the front door, by having the upper portion of the cranked pillar vertical, thus permitting the use of a full-length wind-down glass in the window.

Insulation and corrosion prevention

During the past few years, developments have taken place in engines, transmission and exhaust systems that have done much to make even the low-price cars adequately quiet-running. However, it has been up to the owner to deal with many of the problems of rust attack and the inherent drumming associated with metal box-like structures. Examination of the latest products reveals that most of the manufacturers are at last making real progress in the sealing and sound-deadening of bodywork during manufacture.

As an example of this better outlook, the Citroën DS.19 has a thick layer of insulating material covering the underside of the bonnet. Underneath the huge bonnet of the Pontiac Bonneville is a large sheet of insulation, approximately ¾ in thick, which is held in position by lengths of thin wire secured in small clips. This method of securing the insulation ensures the minimum transmission of noise to the metal panel through the insulation attachment fittings. Another feature of the bonnet and boot lids of several of the new models is that their undersides are treated with an insulating compound as a means of preventing drumming.

The facia of the Panhard Dyna is covered in the body trim material

The rear-end styling of the Dodge Custom Royal is emphasized by the projecting form of the lamps and of the bumper over-riders

In addition to the treatment of such external panels, there is an increasing tendency to use thicker underfelt than before inside the car. This is secured underneath carpets, around propeller shaft tunnels, on the forward toeboards and in the rear seat pan, near the rear axle. In addition, many cars now feature carpets with thicker piles than previously; this of course is also a sound-deadener. The roof of the Simca P60 Aronde is fitted with a layer of sound-insulating material, as is the Dutch DAF small car. The headlining of the DAF car is held in position in the roof by a series of metal pressings adjacent to the cant rail, instead of by the more usual method of fastening it to roof-sticks and bringing it down the sides of the roof in a sweep.

The box sections necessary on modern all-metal bodywork, particularly in the floor area, are traps for moisture, which attacks the inside of the sections and ultimately rusts them away. Many of the all-metal body structures exhibited this year are dipped successively into large baths of rust-proofing solution and paint, to ensure that all interior and exterior surfaces are proofed against rusting for several years of service life. This treatment is undertaken after the main components of the body shell have been assembled, and includes the complete immersion of the lower part of the body in a bath of paint, followed by stoving to provide a hard surface. Later in the process of manufacture, the whole floor is sprayed with an anti-drumming compound.

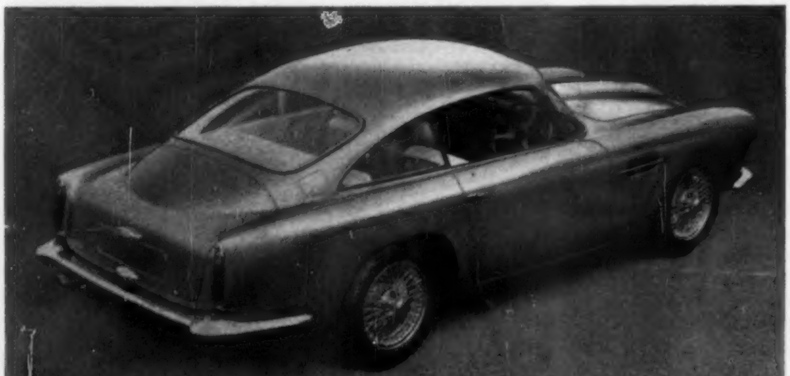
Interiors

The rapidly improving quality of interior trim and furnishing on the more popular cars is very marked. For reasons of economy, it cannot be expected that the use of high-quality materials such as figured wood for facias and door cappings, and leather for interior trimming, should be standard in all family saloon cars, but it is a sign of the times that these materials are no longer confined to specialist body production.

Of the recently announced cars, the Rover 3 Litre is an example of quality furnishing. Its deep seats are styled with a wide roll along the front and sides of the cushions, together with a series of flutes across the mid-portion, both on the cushions and the squabs. A walnut finish has been selected for the glove locker lids and for the door fillets; these fillets take the form of strips just below the level of the windows.

On the Humber Super Snipe, burr-walnut veneer is employed for the complete facia and for the wider door cappings, and also for the two folding tables provided for rear seat passengers. The Jaguar Mark IX has a similar interior finish of quiet dignity and, in addition, is provided with a lockable compartment on the back of the front seat. Leather trimming is featured on the Armstrong Siddeley Star Sapphire, and a two-colour trim is obtainable—black for the outer roll of the seats and squabs, and white for the seating portion. In the lower-price range, the Wolseley Fifteen-Hundred is provided with a walnut veneer on the facia and door cappings.

A safety feature that is becoming increasingly popular is the use of a shock-absorbent, moulded roll across the whole of the front compartment, just above the facia panel. This safety moulding, usually in a relatively hard rubber-like material, in some instances covers the space between the lower inside edge of the windscreen and the top of the facia, and terminates in a deep roll section protruding over the edge of the facia. The Standard Vanguard is fitted with a padded roll below the facia panel instead of in the more usual position above it. In the case of the Swedish Volvo



Unorthodox body construction methods are employed on the Aston Martin DB4, which has highly aerodynamic lines

The new Rover 3 Litre car has unitary construction of chassis and body. A two-tone colour scheme emphasizes the waist-line

122S, the safety roll is extended, with tapering section, for some distance along each front door; on the Rover 3 Litre, the roll is continued along the full length of the doors, underneath the windows; while on the Bristol 406, rolls are provided above and below the facia. In the Vauxhall Velox and Cresta models, the moulded safety roll is covered with dark leathercloth and so placed that reflections from a high sun are not reflected back from it into the driver's eyes.

On the Volvo 122S, the facia safety roll is taken along the forward portion of the doors

Another interior feature that has definitely once again increased in popularity with the introduction of the latest models is the provision of map pockets. The type and location of these varies: on the Panhard Dyna, for example, there is a triangular-shape pocket, in soft material, mounted on the trim panel and fitted with an expanding elastic top. This type of pocket was very popular in pre-war cars, but it was found that ultimately the top sagged, because of the quantity of items crammed in it, and soon became unsightly. To avoid this trouble, several different designs are now used. One such device is a hardboard pocket, hinged at the bottom of each front door and held close to the trim panel by strips of elastic passed through the trim and secured behind it. This type is employed on the Rover 3 Litre saloon and several other models.

On the backs of the front seats of the Daimler Majestic are two pleated map pockets; additional pocket space is provided in all four doors, by using some of the space between the outer panel and the door trim panel. The Standard Vanguard has a spring-loaded netting map-carrier, on the scuttle panel, beneath the facia, on the driver's side. On the Bristol 406, the space above the rear wheel arches, which is normally lost, has been put to good use by incorporating a pocket between the rear seat armrests and the outer panel. This design gives plenty of room in the space above the pocket for the elbows of the rear passengers. Another ingenious method of providing large pockets is used on the Mercedes-Benz 190SL. Each door trim panel is fitted with an elongated armrest, which actually surrounds the top of a deep pocket, set in the space between the outside panel and the trim.

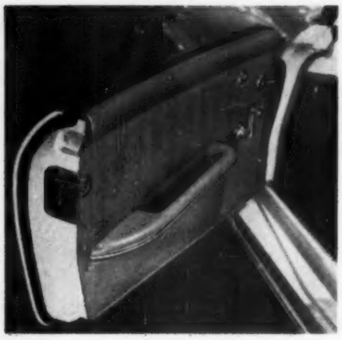
Most saloon bodies have a parcels shelf behind the rear seats, but there is a noticeable increase in the use of some form of parcels tray beneath the facia panel. Designs range from a tray across the full width of the body, embodying a small vertical flange to prevent items from slipping on to the floor, to a half-tray on the passenger's side only, tilted well back to form a rack particularly suitable for papers. On the passenger's side of the body of the Volvo 122S there is a small corner tray, having a ledge round the two open edges.

The grouping of instruments on a panel above the steering wheel, and the provision of a shroud to avoid reflections when the instruments are illuminated, is found on quite a number of the 1959 cars. A concentration of the instruments and, as far as possible, the switches and controls, into a small area, leaves the passenger's side of the car clear and at the same time facilitates the employment of a compact sub-assembly, which is convenient for fitting when the body is manufactured. This has been a feature of the current range of Ford cars since their inception, and it is used on most of the high-speed sports and touring cars.

Since leather suitable for interior trim has become more

To give an unobstructed boot on the Armstrong Siddeley Star Sapphire, the spare wheel and tools are carried in a separate compartment

readily available, it has taken the place of much of the leather-cloth previously employed on seat cushions and squabs, and on door trim panels of the more expensive quantity produced cars. Nylon as a material for interior trim is also being increasingly used. A most attractive nylon cord fabric is offered, as an alternative to leather, for the trim of the 1959 Vauxhall Cresta models, and two nylon fabrics are used on the Ford range of cars: a Jacquard design named Boodle—a mixture of cotton and nylon—and a material called Diamond—a mixture of nylon and viscose, producing a spot effect—are available as alternatives to hide on the Ford Zodiac and the de luxe Consul. Either type of material is available at extra cost on the Consul and Zephyr saloons, and the Diamond material is available on the Anglia and Prefect saloons, also at extra cost, to customers who want it.



The Mercedes-Benz 190SL features rigid pockets in the door trim; they are topped by arm rests

On the Rover 3 Litre, rain running down the screen pillar is caught by a drain channel. Note the wide parcels shelf below the facia

These nylon fabrics are hard-wearing, by virtue of the fact that nylon has a very high resistance to abrasion. They are easy to produce, and offer to the interior stylist a range of strong, clear colours or pastel shades according to requirements. The material is smooth, so the driver and passengers can move across it easily when getting in and out of a car, but it is claimed that it is not so smooth that they slide during fast cornering. In the Jaguar Mark VIII and Mark IX models, nylon is used for the luxurious pile carpets, giving hard-wearing qualities coupled with ease of keeping the carpets clean and in good condition.

Arrangement of centre armrests varies from car to car. Some have a folding armrest in the middle of the rear seats, while others have it in the middle of the front seat; again, one or two cars have both seats so equipped. Reclining front seats are fitted to a number of cars and those on the Bristol 406 have a hinge-up type of headrest above the squab of each of the separate seats. In styling and construction, seats are becoming more like those of aircraft, in that deep rolls are more widely employed on the forward edges, and the trim is fluted either fore and aft or across the seats, according to the effect desired. In the rear of the Peerless G.T. coupé, a very deep leg roll is provided. The seat and squab are formed by a single polyester-glass moulding.

Interiors of the exhibits of all classes of cars show the importance that manufacturers now give to colours and their application to interior trimming. The use of two colours of material for seat and trim is widespread, and the shades are selected to harmonize with the exterior finish. Not only are materials carefully selected for the passenger portion of the body, but also suitable material is applied to the interior of the boot, thereby assisting in the insulation and preventing irritating rattles. At the same time, such covering protects trunks and cases from being damaged due to movement.

There is still ample scope for utilizing parts of the car bodywork for items such as flasks, which cannot easily be accommodated in glove boxes or pockets. Such items, when placed on the rear parcel shelf, can be a nuisance to the driver because they obstruct vision to the rear. It is also useful to have them somewhere convenient to the driver's hand. A novel feature for enhanced passenger comfort is the up and down adjustment of the outer armrests, as fitted to the Rover 3 Litre saloon. By pressing a recessed button in the centre of the armrest, a catch is released at the back of the rest and the whole unit can then be raised or lowered.

NEW STANDARDS

IN 1952, Part 1 of the revised British Standard 731 was published under the title of "Flexible steel conduit and adaptors for the protection of electric cable". The second part of this Standard, an illustrated eight page publication, is now available; it is entitled "Flexible steel tubing to enclose flexible drives for power-driven tools for general purposes". It deals with the interlocking and locking types of flexible tubing and specifies dimensions and mechanical requirements as well as suitable manufacturing materials.

The latest in the B.S.I. series of process standards on welding is B.S. 2996 : 1958, which deals with the projection welding of low-carbon wrought steel studs, bosses, bolts, nuts, and annular rings to plate, sheet, strip and tube. This illustrated standard, of twelve pages, lays down the requirements for the following: the material and design of the electrodes and their inserts; welding procedure and weld inspection; and testing details and the results that should be achieved. An appendix lists a number of operating instructions relevant to both pedal-operated and power-operated machines.

B.S. 731, Part 2 : 1958 and B.S. 2996 : 1958 are both obtainable from the British Standards Institution, Sales Branch, 2 Park Street, London, W.1. The first is priced at 3s and the second at 4s; postage is charged extra.

Mounting Anti-Friction Bearings

A Study of the Reduction of Diametric Clearance in Bearings Mounted with an Interference Fit

FOR a number of bearing applications it is often of interest to calculate the extent of the reduction in internal diametric clearance which occurs as a result of the bearing being mounted with an interference fit on the shaft and in the housing. So that the magnitude of the reduction in clearance can be readily evaluated for different diameter ratios (see below) of shafts (hollow shafts) and housings, graphs have been prepared which show the relationship between the change in track diameter of the bearing rings and the effective interference.

The graphs apply to pressure joints of the type shown in Fig. 1, consisting of an outer and an inner component of the same width. The components can be considered as narrow, and therefore the axial stresses may be disregarded and the stresses existing in the transverse radial plane only need be taken into consideration.

The following symbols are used:—

E = modulus of elasticity

$\frac{1}{m}$ = Poisson's ratio

d, D = diameter of fitting surface

c = diameter ratio $= \frac{\text{inside diameter of component}}{\text{outside diameter of component}}$

p = pressure at fitting surface

Δ = effective interference between the components

δ = change in diameter of free surface of component

T = tolerance

Suffixes:—

i = inner component of joint

e = outer component of joint

In the elastic range of the material the following relationship exists between the interference Δ and the pressure p.

$$\frac{\Delta}{d} = \frac{p}{E_e}\left(\frac{1+c_e^2}{1-c_e^2}+\frac{1}{m_e}\right)+\frac{p}{E_i}\left(\frac{1+c_i^2}{1-c_i^2}-\frac{1}{m_i}\right) \quad \ldots\ldots\ldots\ldots(1)$$

The following applies to the increase in the outside diameter of the outer component of the joint

$$\frac{\delta_e}{d} = \frac{2p \cdot c_e}{E_e(1-c_e^2)} \quad \ldots\ldots\ldots\ldots(2)$$

Similarly, the following applies to the reduction in the inside diameter of the inner component of the joint

$$\frac{\delta_i}{d} = \frac{2p \cdot c_i}{E_i(1-c_i^2)} \quad \ldots\ldots\ldots\ldots(3)$$

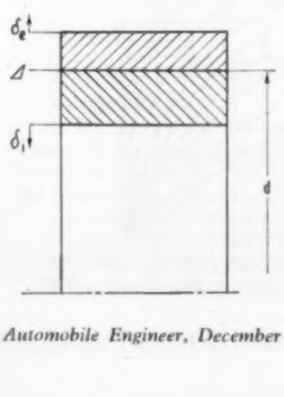


Fig. 1. Pressure joint consisting of two components having the same width

Dividing equation (2) by equation (1) yields the following for the outer component of the joint

$$\frac{\delta_e}{\Delta} = \frac{\dfrac{2c_e}{E_e(1-c_e^2)}}{\dfrac{1}{E_e}\left(\dfrac{1+c_e^2}{1-c_e^2}+\dfrac{1}{m_e}\right)+\dfrac{1}{E_i}\left(\dfrac{1+c_i^2}{1-c_i^2}-\dfrac{1}{m_i}\right)} \quad \ldots\ldots\ldots\ldots(4)$$

For the general case, where the inner and outer components of the joint are made of the same material, i.e. $E_i = E_e$ and $\frac{1}{m_i} = \frac{1}{m_e}$, this equation is simplified to

$$\frac{\delta_e}{\Delta} = \frac{\dfrac{2c_e}{1-c_e^2}}{\dfrac{1+c_e^2}{1-c_e^2}+\dfrac{1+c_i^2}{1-c_i^2}} \quad \ldots\ldots\ldots\ldots(5)$$

which *applies to the outside diameter of the bearing inner ring*. In this instance Δ is the effective interference between the bearing inner ring and the shaft.

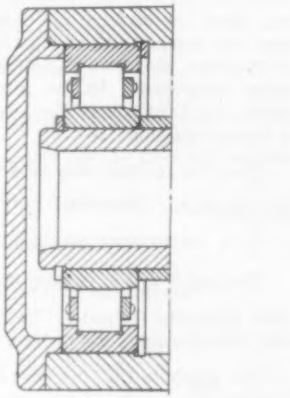


Fig. 2. Bearing scheme with cylindrical roller bearing on hollow shaft

Dividing equation (3) by equation (1) yields the following for the inner component of the joint

$$\frac{\delta_i}{\Delta} = \frac{\dfrac{2c_i}{E_i(1-c_i^2)}}{\dfrac{1}{E_e}\left(\dfrac{1+c_e^2}{1-c_e^2}+\dfrac{1}{m_e}\right)+\dfrac{1}{E_i}\left(\dfrac{1+c_i^2}{1-c_i^2}-\dfrac{1}{m_i}\right)} \quad \ldots\ldots\ldots\ldots(6)$$

The following simplification is obtained in this case if $E_i = E_e$ and $\frac{1}{m_i} = \frac{1}{m_e}$

$$\frac{\delta_i}{\Delta} = \frac{\dfrac{2c_i}{1-c_i^2}}{\dfrac{1+c_e^2}{1-c_e^2}+\dfrac{1+c_i^2}{1-c_i^2}} \quad \ldots\ldots\ldots\ldots(7)$$

which *applies to the inside diameter of the bearing outer ring*. In this instance Δ is the effective interference between the bearing outer ring and the housing.

Since equations (5) and (7) are of similar form, the suffix in the numerator being appropriate for either inner or outer

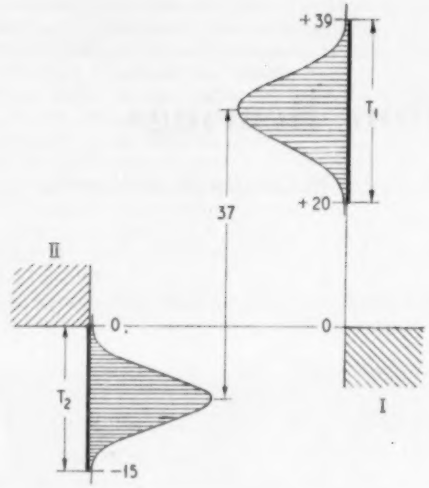


Fig. 3. Diagram showing tolerances for shaft (I) with diameter 60 n6 and bearing inner ring (II)

members, both functions can be represented graphically by one and the same graph.

When determining the reduction in bearing clearance the expansion of the inner ring on the shaft and the contraction of the outer ring in the housing can therefore be read off from the same graph, provided that the bearing, housing and shaft are made of the same material. In the former case the bearing ring is regarded as the outer component of the joint shown in Fig. 1 and in the latter case as the inner component. In the special cases of either a solid shaft ($c_i = 0$) or a very rigid housing ($c_e \approx 0$), equation (5) is further simplified to

$\frac{\delta_e}{\Delta} = c_e$ for an inner ring on a solid steel shaft ....(8)

and equation (7) becomes

$\frac{\delta_i}{\Delta} = c_i$ for an outer ring in a very rigid steel housing ..(9)

The graph in Fig. 4 gives the function $\frac{\delta}{\Delta}$ in accordance with equations (5) and (7) for both bearing rings, assuming that both the shaft and the housing are made of steel.

The graph in Fig. 5 gives the function $\frac{\delta_i}{\Delta}$ in accordance with equation (6) for a bearing outer ring in a bearing housing made of cast iron with $E = 15 \times 10^6$ lb/in² and $\frac{1}{m} = 0{\cdot}25$.

Finally, the graph in Fig. 6 gives $\frac{\delta_e}{\Delta}$ in accordance with equation (4) for a bearing inner ring on a cast-iron shaft with $E = 15 \times 10^6$ lb/in² and $\frac{1}{m} = 0{\cdot}25$.

Where steel components are involved, the modulus of elasticity and Poisson's ratio have been taken as $E = 30 \times 10^6$ lb/in² and $\frac{1}{m} = 0{\cdot}3$.

The following additional remarks apply to the validity of the formulae:

The diameter ratio for a bearing ring is determined on the basis of a hypothetical ring of rectangular section, which has the same width and sectional area as the actual bearing ring.

Generally, the bearing rings are narrower than the shaft and the housing, and the values obtained from the graphs therefore do not correspond exactly with the actual conditions. In addition the bearing housings are, for example, often made more rigid by flanges and closures. In each case, therefore, due allowance must be made for the effect of this stiffening on the change in the diameter of the bearing ring.

Smoothing of the contact surfaces of a pressure joint always occurs due to the micro-geometrical surface differences being subjected to elastic and plastic deformation, and the effective interference will be less than that obtained from the dimensions measured before mounting. If the surface finish is not known, the following typical values for the compression of the surface layer due to smoothing of the mating surfaces may be used, depending on the type of machined finish:

Fine-ground surface	1–2·5 μ
Fine-turned surface (high speed and fine feed) ..	3–7 μ
Machine-reamed bore	5–12 μ
Fine-turned surface (slow speed)	12–24 μ

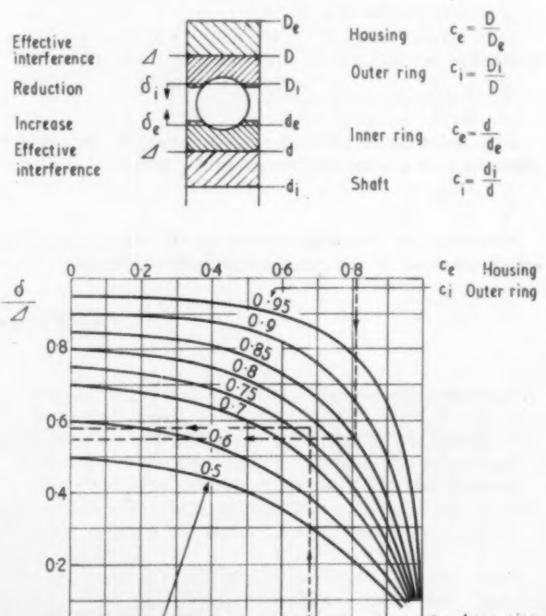
If, on the other hand, the surface finish is known, approximately half the profile depth may be taken as the value for the smoothing.

The smoothing of the bearing ring may be taken as 1–2 μ.

The following example shows how the graphs are used. When mass-produced components are used in the joint the theoretical limits of the interference are less likely to occur, and the example takes into account the probable variation in interference and the probable limit values.* It is therefore assumed that for a large number of such components, the diameter values of the fitting surfaces are distributed round a mean value in accordance with normal distribution curves.

*See article "The probable interference in ball bearing fits", published in Ball Bearing Journal No. 3, 1951. (Skefko Ball Bearing Co. Ltd.)

Fig. 4. Change of outside diameter of inner ring and inside diameter of outer ring in relation to the effective interference, $\frac{\delta}{\Delta}$, with bearing mounted on a steel shaft and in a steel housing



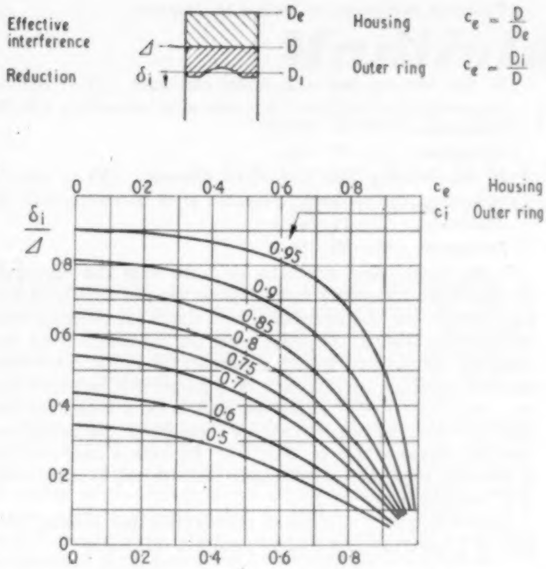


Fig. 5. Change of inside diameter of outer ring in relation to the effective interference, $\frac{\delta_i}{\Delta}$, with bearing mounted in a cast-iron housing ($E = 15 \times 10^6$ lb/in², $\frac{1}{m} = 0{\cdot}25$)

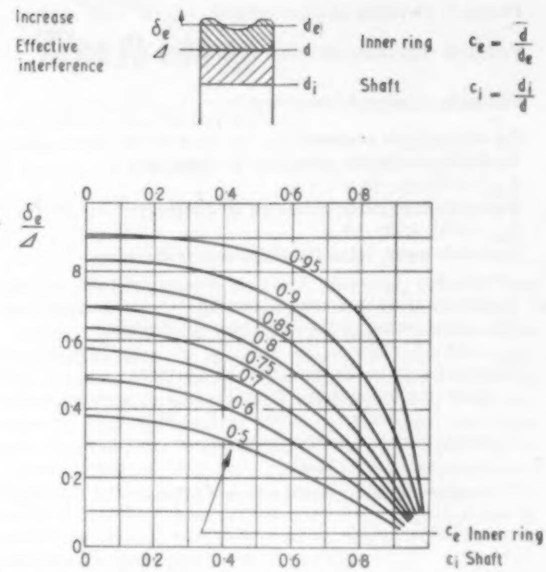


Fig. 6. Change of outside diameter of inner ring in relation to the effective interference, $\frac{\delta_e}{\Delta}$, with bearing mounted on a cast-iron shaft ($E = 15 \times 10^6$ lb/in², $\frac{1}{m} = 0{\cdot}25$)

All dimensions are in millimetres in the example; 1 μ = 0·001 mm.

Example: A cylindrical roller bearing NU 312/C3 with dimensions 60×130×31 mm is to be mounted on a hollow shaft and in a bearing housing which has no appreciable stiffening, see Fig. 2. The maximum clearance in the unmounted bearing is taken as 75 μ and the minimum as 55 μ. It is assumed that an interference fit is used for both bearing rings; n6 limits apply to the shaft diameter and N7 limits to the diameter of the bearing housing, see Fig. 3. All the components are made of steel. For the bearing scheme concerned it is known from experience that a minimum bearing clearance of 10 μ is required after mounting.

1. *Increase in diameter of track of inner ring* (Assume $d_i = 41$ mm; $d_e = 77$ mm)

The diameter ratio of the hollow shaft $c_i = \frac{41}{60} = 0{\cdot}68$

The diameter ratio of the inner ring $c_e = \frac{60}{77} = 0{\cdot}78$

The graph in Fig. 4 gives the increase in the outside diameter of the inner ring in relation to the effective interference, $\frac{\delta_e}{\Delta} = 0{\cdot}58$

Shaft diameter 60 n6. Limits $\begin{matrix} +39\ \mu \\ +20\mu \end{matrix}$

Tolerance $T_1 = 19\ \mu$

Inner ring bore 60 KB. Limits $\begin{matrix} -15\ \mu \\ 0\ \mu \end{matrix}$

Tolerance $T_2 = 15\ \mu$

For the example selected a smoothing of 1 μ is used for the surface of the bearing ring and 2 μ for that of the shaft. Owing to smoothing, the reduction in interference will therefore be $2(1+2) = 6\ \mu$.

The following is therefore obtained:
Minimum effective interference $20 - 6 = 14\ \mu$
Maximum effective interference $54 - 6 = 48\ \mu$
Mean effective interference 31 μ

Probable variation in interference
$= \sqrt{T_1^2 + T_2^2} = \sqrt{19^2 + 15^2} = 24\ \mu$

Probable minimum interference $= 31 - \frac{24}{2} = 19\ \mu$

Probable maximum interference $= 31 + \frac{24}{2} = 43\ \mu$

The following is thus obtained for the track of the inner ring:
Probable minimum increase in diameter
$\delta_{min} = 0{\cdot}58 \times 19 = 11\ \mu$
Probable maximum increase in diameter
$\delta_{max} = 0{\cdot}58 \times 43 = 25\ \mu$
Probable mean value for increase in diameter
$\delta_m = 18\ \mu$

2. *Reduction in diameter of track of outer ring* (Assume $D_e = 160$ mm; $D_i = 111$ mm)

The diameter ratio of the housing $c_e = \frac{130}{160} = 0{\cdot}81$

The diameter ratio of the outer ring $c_i = \frac{111}{130} = 0{\cdot}85$

The graph in Fig. 4 gives the reduction in the inside diameter of the outer ring in relation to the effective interference, $\frac{\delta_i}{\Delta} = 0{\cdot}55$.

Housing bore 130 N7. Limits $\begin{matrix} -52\ \mu \\ -12\ \mu \end{matrix}$

Tolerance $T_1 = 40\ \mu$

Outer ring diameter 130 hB. Limits $\begin{matrix} 0\ \mu \\ -18\ \mu \end{matrix}$

Tolerance $T_2 = 18\ \mu$

It is assumed that the smoothing of the fitting surface in the bearing housing is 4 μ. Due to the smoothing of the surfaces, the reduction in interference will thus be $2(1+4) = 10\ \mu$.

Minimum effective interference $-16\ \mu$ (clearance fit)
Maximum effective interference 42 μ (interference fit)
Mean effective interference 13 μ

Probable variation in interference $=\sqrt{40^2+18^2}=44\ \mu$

Probable minimum interference $=13-\frac{44}{2}=-9\ \mu$

Probable maximum interference $=13+\frac{44}{2}=35\ \mu$

The following is obtained for the track of the outer rings:

Probable minimum reduction in diameter

$\delta_{min}=0{\cdot}55\times(-9)=-5\ \mu$

Probable maximum reduction in diameter

$\delta_{max}=0{\cdot}55\times35=19\ \mu$

Probable mean value for reduction in diameter

$\delta_m=7\ \mu$

3. *Reduction of clearance in the bearing*

The mean value of the reduction in clearance

$=18+7=25\ \mu$

The variation in the reduction in clearance

from the inner ring $=25-11=14\ \mu$

„ „ outer „ $=19-(-5)=24\ \mu$

Probable variation in the reduction in clearance

$=\sqrt{14^2+24^2}=28\ \mu$

Probable minimum reduction in clearance

$=25-\frac{28}{2}=11\ \mu$

Probable maximum reduction in clearance

$=25+\frac{28}{2}=39\ \mu$

If the bearing has maximum clearance ($75\ \mu$) before mounting, the probable clearance after mounting will be

Maximum $=75-11=64\ \mu$

Minimum $=75-39=36\ \mu$

If the bearing has minimum clearance ($55\ \mu$) before mounting, the probable clearance after mounting will be

Maximum $=55-11=44\ \mu$

Minimum $=55-39=16\ \mu$

If the production methods are such that the specified tolerances for the shafts and housings are not fully utilized, but, instead, the diameters of all the shafts are situated near the top limit and, at the same time, the diameters of all the housings are situated near the bottom limit, the minimum residual bearing clearance will be considerably less than the $16\ \mu$ obtained above. If the top and bottom limits for the shaft and housing respectively are considered, the minimum residual clearance will be only $7\ \mu$. In such a case the use of bearings with internal clearance towards the bottom limit of C3 should be avoided.

Acknowledgment is made to the Skefko Ball Bearing Co. Ltd. of Luton, for permission to publish this study.

INSTITUTION OF MECHANICAL ENGINEERS

Meetings of the Automobile Division

Birmingham

Thursday, 11th December, 6.30 p.m. at the James Watt Memorial Institute, Great Charles Street, Birmingham. Joint Meeting with Midland Branch.

Coventry

Tuesday, 6th January, 7.15 p.m. in the Grosvenor Room, Leofric Hotel. Three Papers on "I.C. Engine Dynamics", by R. Clink, C. A. Beard, A.M.I.Mech.E., and D. R. Braund, A.M.I.Mech.E.

Derby

Monday, 15th December, 7.15 p.m. at the Midland Hotel, Derby. Automobile Division Chairman's Address entitled "Years of Development", by R. A. Wilson-Jones, M.I.Mech.E.

Monday, 19th January, 7.15 p.m. at the Midland Hotel, Derby. Paper: "Air Suspension for Road Vehicles", by J. H. Sainsbury, A.M.I.Mech.E.

Luton

Wednesday, 10th December, 7.30 p.m. in the Assembly Room, Luton Town Hall. Paper: "Standards and Standardization in the Motor Industry", by E. W. Woodbridge.

Wednesday, 7th January, 7.30 p.m. in the Assembly Room, Luton Town Hall. Automobile Division Chairman's Address: "Years of Development", by R. A. Wilson-Jones, M.I.Mech.E.

North-Eastern

Wednesday, 17th December, 7.30 p.m. in the New Chemistry Theatre, The University, Leeds. Paper: "Air Suspension for Road Vehicles", by J. H. Sainsbury, A.M.I.Mech.E.

Scottish

Monday, 15th December, 7.30 p.m. at the Institution of Engineers and Shipbuilders, Glasgow. Technical Film Show.

Monday, 19th January, 7.30 p.m. at the Institution of Engineers and Shipbuilders, Glasgow. Lecture: "The Working of Sheet Metal and its Fabrication for Motor Bodies", by G. R. Allen.

ESSAY COMPETITION

TO encourage the study of diesel engines and their maintenance, a new essay competition has been announced by the Institute of the Motor Industry. It is open to registered students of technical colleges, and two awards are offered by Leslie Hartridge Ltd., of Buckingham; the first prize is fifteen guineas and the second ten guineas. The subject for this competition is "The Servicing of Diesel Fuel Pumps and Injectors".

Entries should exceed 1,000 words and must reach the Secretary of the Institution not later than 1st April, 1959. Full details have been circulated to all technical colleges. Prizewinners will be given the opportunity of spending a day at the Hartridge factory, with expenses paid, to obtain first-hand knowledge of the design and manufacture of fuel pump servicing equipment.

TECHNICAL INFORMATION

WITH the post-war technological advances in so many directions, it is becoming increasingly important that technical information should be expressed concisely, accurately and comprehensively. On this account, the Manchester College of Science and Technology is organizing a course of eight lectures entitled "The Presentation of Technical Information". The course will commence on Tuesday, 17th February, 1959 at 7 p.m.; there will be two two-hour lectures, on Tuesdays and Wednesdays, on that and the three following weeks.

Enrolment for the course will take place during the half hour preceding the first lecture, when the course fee of £1 10s will be payable; no charge will be made to associates of the college. Further details and application forms may be obtained, on request, from the college.

Manifold Tuning

Reassessment of the Basic Principles and Potentialities

By E. M. Goodger, M.Sc., Ph.D., A.M.I.Mech.E., A.F.R.Ae.S., F.Inst.Pet.,

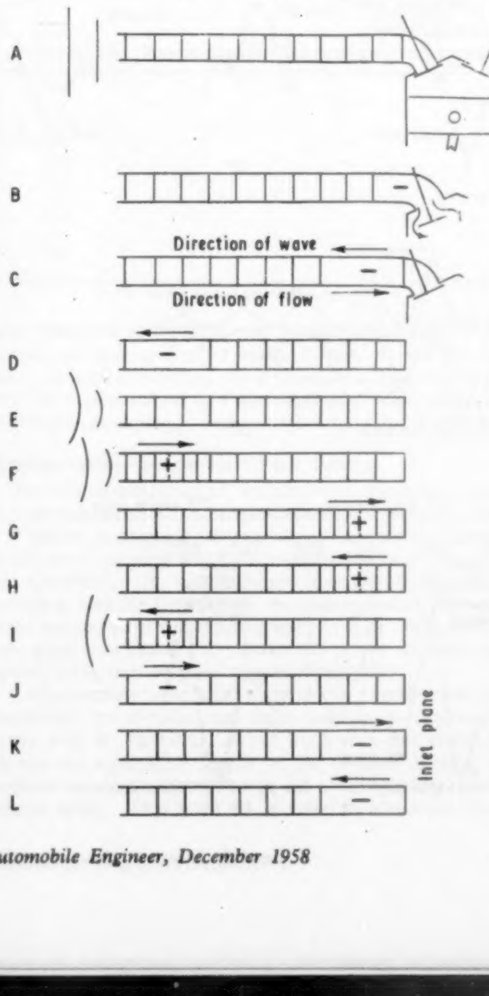
and J. Hillsdon, D.C.Ae., G.I.Mech.E., Grad.R.Ae.S.

Many devices are marketed for the improvement of engine power, economy, anti-knock requirement, life, and so on. These improvements must be assessed eventually against the initial cost of the device, and the expense of its operation and maintenance. All costs will be kept down if the device is simple, and if existing forces can be utilized.

There are obvious disadvantages to the conventional system of supercharging for increased power output, where a positive pressure is maintained continuously at the inlet ports by means of an engine-driven compressor. However, some degree of free supercharging is found to be possible in a normally-aspirated engine, by matching the induction processes in the engine cylinder with the natural vibrations of gas pressure set up in the inlet manifold. Some depression in the manifold is inevitable during induction, but a positive pressure pulse arriving at the instant of valve closure will ram in the last portions of the charge and ensure maximum charge density. Synchronism is required, therefore, between inlet valve operation and the pressure pulsations in the manifold.

In theory, this synchronism is obtainable with no more complication than a specified length of manifold pipe. Unfortunately, the requisite pipe length is found to vary with engine speed and, in practice, these lengths may be inconvenient. Furthermore, the necessity for manifold bends, carburettor chokes, hot-spots, throttles and air filters, and the relative timing of induction strokes with a multi-cylinder engine all tend to confuse the issue. The purpose of this article is to review the requirements with regard to manifold length for tuning purposes, and to assess the extent of performance gains obtainable in practice.

Fig. 1. The valve closed condition is shown at A; when the valve opens, as at B, the depression generated by the induction stroke is propagated along to the open end of the manifold, as shown at C and D. From there, it is reflected as a pressure pulse, as at E and F, which returns to the closed end G. Then the pulse reflects, as shown at H, I, J, K and L



Pressure pulses in a tube

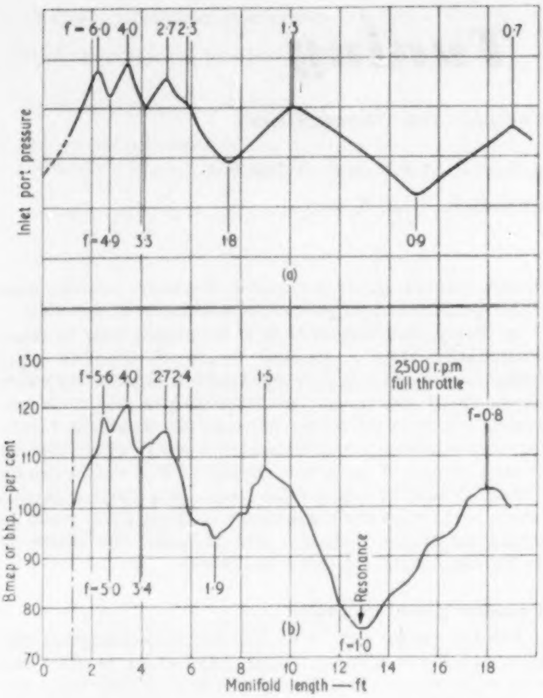
For the greater part of its full-throttle working life, the inlet manifold of a single-cylinder engine exists, in effect, as a tube with one end closed by the inlet valve and the other end open to atmosphere, as shown at A in Fig. 1. When a depression is created at the valve end of the manifold, by the induction stroke of the piston, which is represented by condition B in Fig. 1, the depression propagates along to the open end of the manifold as shown at C and D. From there, the depression is reflected as a pressure pulse, E and F, which propagates back to the closed end, as at G, to be reflected once more as a pressure pulse, H. At the open end, the pulse changes sign again and returns as a depression, I and J. Then the initial condition is regained, K, and the processes repeated, L and onwards.

Throughout this process, pressure losses ensue owing to the pulse reversals and to friction, and the wave amplitude attenuates. Hillsdon[1], from consideration of results presented by Helmholtz, and by Vautier, de la Boulaye, and Balme, shows this attenuation loss to be equal to approximately 20 per cent. In the case of a very long manifold, therefore, the pressure changes at the inlet plane would appear as in Fig. 2a. With a shorter manifold, the pressure waves would overlap, as in Fig. 2b, and only the resultant pressure variation would be experienced. In both cases, the formation of minor pulses, due to gas inertia, at the inlet plane after condition B, has been ignored. The magnitude of these pulses would be largely a function of the valve closure rate.

Matching manifold to engine

The normal timings for inlet valve opening and closure in a four-stroke engine are a few degrees before top dead centre, and a greater number of degrees after bottom dead centre, respectively. Since only minor flows are incurred at the beginning and end of the valve operating period, the effective timings of opening and closure may be assumed to occur at top and bottom dead centre respectively; this will simplify the analysis.

The resultant pressure curve of Fig. 2b has been superimposed upon the four-stroke engine-event diagram in Fig. 2c. Since a maximum positive pressure is obtained at the instant of valve closure, the value of $0{\cdot}5\ t_1$ for the reflection time t is an optimum. The relationship between the



manifold pressure pulses and the engine events may be expressed in terms of a frequency ratio, defined as follows:

$$\text{Frequency ratio} = f = \frac{\text{pressure pulse frequency}}{\text{valve opening frequency}}$$

The periodic time of the pressure pulse is equal to 2t, and the frequency is the inverse of this. For a four-stroke engine, the valve opening frequency is N/120, where N is in r.p.m. The general expression for the frequency ratio then becomes:

$$f = \frac{60}{Nt}$$

For the optimum tuned case, represented in Fig. 2c, the reflection time is equal to $0{\cdot}5\ t_1$, and this is equivalent to 90 deg of crank angle rotation, that is $t_{opt} = 60/4N$.

$$\text{Hence, } f_{opt} = \frac{60}{N} \times \frac{4N}{60} = 4$$

Examination of Fig. 2c shows, in addition, that the residual wave from the first engine cycle augments the charging processes of the next cycle. The amplitudes of each successive resultant so formed will rise progressively, to reach a constant value after a few cycles.

A further point illustrated in Fig. 2c is that the residual peak N, at the instant of valve closure in the second cycle, is an exact number of peaks later than the corresponding peak M in the first cycle. Clearly, similar advantages should accrue with other whole numbers of peaks between M and N, although the effects would not be so great since the resultant peak M would be reduced in magnitude, and would move from its optimum position.

In the case of a very long manifold, as represented in

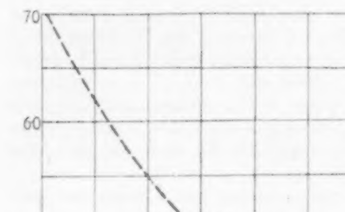


Fig. 5. Above: Curves showing how optimum manifold length varies with engine speed

Fig. 3. Above: Predicted pressure variations in the port are shown at (a), while the experimental values of b.m.e.p. obtained with different lengths of manifold are plotted to form the other curve at (b)

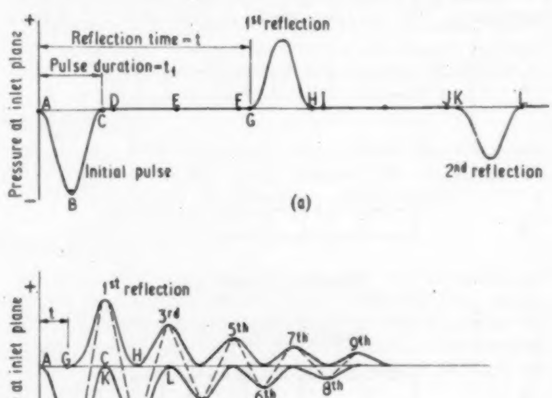


Fig. 2. Right: The time relationship at the inlet plane of a long manifold when $t > t_1$ is shown at (a); when $t < t_1$, in this case $t = 0{\cdot}5t_1$, it is as shown at (b); the optimum condition, when the resultant pressure wave is synchronized with the engine cycle, is that illustrated at (c)

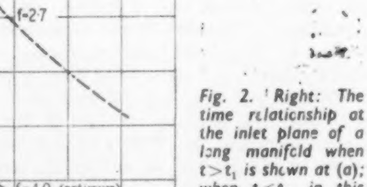


Fig. 4. Right. The manifold arrangement used for the experiments with the Fedden sleeve-valve engine

Fig. 2a, it is conceivable that the reflection could be delayed until an alternate cycle is reached, so that the first reflection from the first cycle augments the pressure at inlet valve closure during the third cycle.

By inspection of the pressure waves for all these cases, it is possible to estimate the appropriate values of t, and their corresponding values of f. These theoretical values for maximum port pressures are listed in the Table. By similar reasoning, the conditions for minimum port pressures at valve closure can also be determined, and these are also included in the Table. Fig. 3a is an arbitrary port-pressure curve, showing the expected peaks and troughs at the theoretical values of f.

The frequency ratio can also be expressed in terms of manifold length, L, and pulse velocity, a. From this data it is possible to determine the optimum length of manifold for a given engine speed. The manifold length is traversed four times during the complete cycle of pressure events.

Hence, $f = \frac{a}{4L} \times \frac{120}{N} = \frac{30a}{L\,N}$

that is, $L = \frac{30a}{f\,N}$ and $L_{opt} = \frac{7{\cdot}5a}{N}$

Mucklow[2] shows that 1,117 ft/sec is a reasonable average value for the pulse velocity at ambient conditions, and that the influence of manifold diameter is small. Hillsdon[1] points out that the inlet air mean velocity is of the order of 40 ft/sec, and is therefore negligible in comparison with the pulse velocity.

Thus, $L_{opt} = \frac{8{,}378 \text{ ft}}{N}$

The theoretical lengths of manifold, required for maximum

VALUES OF FREQUENCY RATIO

Peaks		Troughs	
Theoretical	Experimental	Theoretical	Experimental
6·0	5·4	—	—
—	—	4·9	4·8
4·0*	4·0*	—	—
—	—	3·3	3·3
2·7	2·7	—	—
2·3	2·4	—	—
—	—	1·8	1·9
1·3	1·5	—	—
—	—	0·9	1·0
0·7	0·8	—	—

** The figures marked thus are the optimum values.*

port pressures at valve closure, thus obtained for any engine speed, are independent of engine design. It was felt necessary, instead of plotting these theoretical lengths, to check the theoretical values of f against the corresponding values that were subsequently determined accurately by experiment.

Engine tests: comparison with theory

For the investigation of the effects of manifold tuning on the performance of a single-cylinder four-stroke engine, a Fedden horizontal, sleeve-valve, air-cooled engine, of 0·733 litres capacity and 8 : 1 compression ratio, was chosen at Cranfield. A low-pressure petrol-injection system, spraying into the intake port, was incorporated. Most tests were conducted at full throttle and 13 : 1 air : fuel ratio, with the ignition adjusted for maximum power at each engine speed, using the standard length of manifold.

Unfortunately, the inlet manifold is complicated at the valve end, by division into ducts leading to three separate ports, Fig. 4. However, earlier work with this engine[3] had shown the equivalent length of the divided ducting to be approximately 6 inches, and to be roughly independent of engine speed. This value was derived from a comparison of

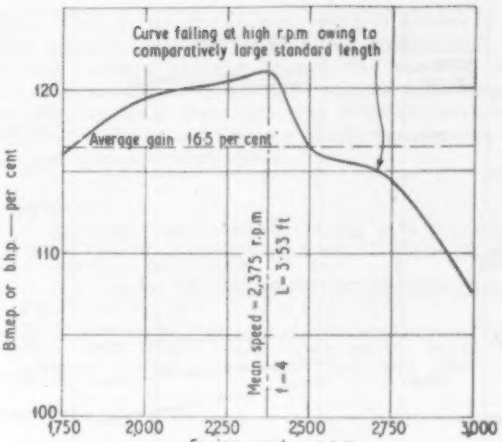


Fig. 6. Curves showing how engine power varies with speed when the inlet manifold has been tuned to give optimum results at mean speed

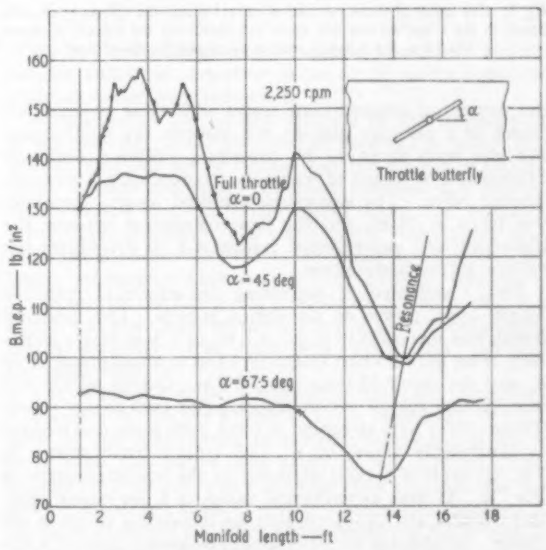


Fig. 7. Above: The effect of throttle setting on the inlet port pressure

Fig. 8. Below: Manifold sections used in tests to determine the adverse influence of curved pipes and elbows on the supercharging effect

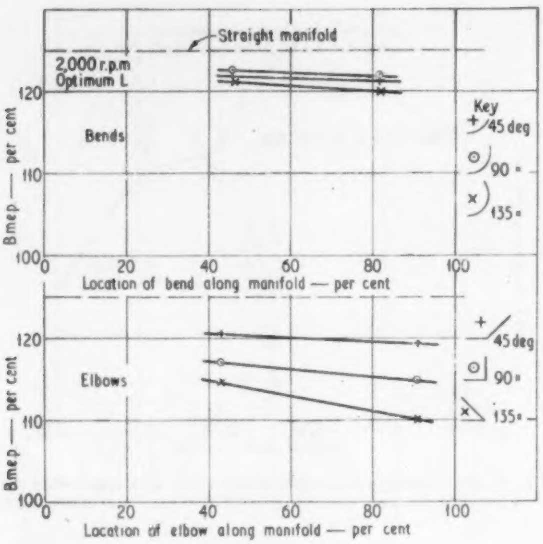
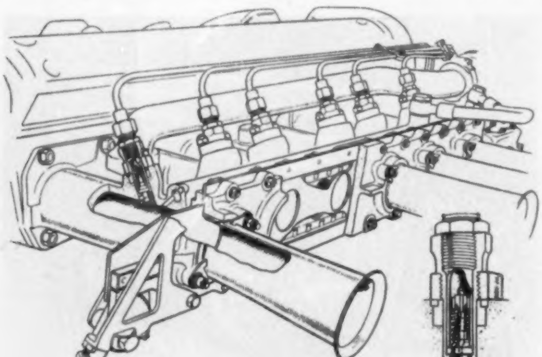


Fig. 9. The upper of these two sets of curves shows the effects of smooth bends in the manifold and the lower set illustrates the effects of sharp elbows on the b.m.e.p. that is developed by the engine

the theoretical pressure-time traces with those obtained by means of a pressure pick-up and cathode ray oscilloscope. For the tests about to be described, Hillsdon fitted an additional 8 in length of manifold, incorporating a butterfly throttle valve. The equivalent standard length, therefore, was taken as 14 in, and the close correlation between the theoretical and experimental results at f = 4 confirms the validity of this assumption.

Test 1 consisted of measuring the effects of manifold length upon power at six engine speeds. The manifold length was increased in steps of 3 in, or 1·5 in where necessary, from the standard length of 14 in to a maximum of 18 ft, and the manifold bore was 1·75 in. Curves of b.m.e.p. were plotted against manifold length, for each engine speed. These curves were all similar in form, with peaks and troughs at comparable values of f. The typical curve shown in Fig. 3b exhibits a close similarity to the predicted curve in Fig. 3a. Average experimental values of f, for power peaks and troughs, are compared with the theoretical values in the Table. In addition to the optimum frequency ratio of 4, the experimental values of f for the other two significant b.m.e.p.

Fig. 10. Manifold tuning can be effectively carried out on the 1956 Jaguar sports car engine equipped with the Lucas fuel injection system



peaks are seen to be 2·7 and 5·4. Lengths of manifold required for maximum b.m.e.p., as determined experimentally, are plotted in Fig. 5.

The extent of power improvement obtained by selecting an optimum manifold length, instead of the standard 14 in length, was found to vary from 13·5 per cent at 3,000 r.p.m., L = 2·79 ft, to 25 per cent at 2,000 r.p.m., L = 4·19 ft. Had the standard length been taken as less than 14 in, these improvements would have been relatively greater, as is demonstrated by the shape of the curves of b.m.e.p. plotted against L.

Inspection of the theoretical pressure waves also indicated the maximum degree of resonance in the manifold at an f value of 0·9. Tests at the corresponding experimental f value of 1·0 showed this to be the case, and the power output was at a minimum.

Engine tests: effect of practical factors

It is considered that, in practice, a manifold of infinitely-variable length, controlled by engine speed, would be impracticable. The objective in the second test, therefore, was to determine the extent of power improvement obtainable by selecting an optimum manifold length for a typical mean speed, and operating the engine over the complete speed range. The results, presented in Fig. 6, show a useful average power improvement of 16·5 per cent, even when operating away from the speed suited to the tuned manifold.

The effect of throttling is shown in Fig. 7, which is based on the third test. When the throttle is at the 45 deg position, the b.m.e.p. peaks are largely eliminated, and when it is 67·5 deg closed, they are completely eliminated. It is interesting to note, however, the persistence of the adverse pressure troughs with the resonant manifold lengths, even at 67·5 deg throttle closure.

Since the results show that, for tuning purposes, appreciable lengths of manifold are involved, the effects of bends and elbows are of direct practical interest. In the fourth test, both smooth bends and sharp elbows were used, and their included angles were 45 deg, 90 deg and 135 deg, Fig. 8. The length of the axes of the 6 in radius bends was 18 in, and that of the axes of the elbows was 9 in. Results for the optimum manifold length at 2,000 r.p.m. are presented in Fig. 9. These show clearly that both bends and elbows reduce the supercharging effect, and that sharp elbows are worse than smooth bends. The reduction generally increases with the angle of bend, but is less when the bend or elbow is located nearer the inlet port. Since the sum of the standard length and the bend axis was 32 in, and the tuned manifold length was 50 in, only two positions were practicable for the bends and elbows with the tuned manifold. Additional tests were conducted with longer manifolds, however, using three positions for the bends and elbows, and similar trends were noted. A few spot tests were made with a smooth bell-mouth entry to the manifold. Although not entirely consistent, the results indicated an additional gain of about 1 per cent in b.m.e.p.

Discussion

The results show that manifold lengths of some 3 to 4 ft are necessary for tuning purposes. It is considered that the conventional manifold layout, comprising one carburettor and multi-offtake stubs, cannot make use of the tuning principle, since the pressure pulses from one port will interfere with and attenuate those from neighbouring ports. Individual manifold pipes are essential and, since multi-carburettors are undesirable, tuning can best be applied to fuel-injection engines. The 1956 Jaguar sports car engine equipped with the Lucas fuel injection system, Fig. 10, is an example of a potential case for manifold tuning. This principle might also be applied to exhaust manifolds to effect super-scavenging. Alternatively, where tuning for power

augmentation is not feasible, it would at least be possible to make investigations to determine what lengths of manifold would be positively adverse in their effects, and then these could be avoided.

Conclusions

An appreciable improvement in performance is obtainable at a given engine speed by tuning the inlet manifold length. A worthwhile improvement in engine performance is obtainable over a typical working range of engine speed by tuning the inlet manifold length to suit the mean speed. Closure of the throttle butterfly progressively eliminates the beneficial pressure pulsations. Smooth bends are preferable to sharp elbows in the manifold, and they should be located as close as possible to the inlet port. Manifold tuning is best suited to fuel-injection engines with individual manifold pipes. This, as has already been stated, is because of the clean design of induction piping that is possible with fuel injection systems, and also because the manifolds used currently on practically all quantity-produced engines are not only too complex in shape for tuning in the way described, but also are of such a design that interference between one port and another is unavoidable.

References

1. J. Hillsdon: "Inlet Manifold Tuning in a Four-Stroke Single-Cylinder Spark-Ignition Engine", *College of Aeronautics Thesis*, June, 1958.
2. G. F. Mucklow: "Exhaust Pipe Effects in a Single-Cylinder Four-Stroke Engine", *I.Mech.E. Proc.* 1940.
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Water-Cooled Brakes

An American System Designed to Reduce Maintenance and Increase Braking Efficiency

WATER-COOLED brakes have recently been used experimentally on aircraft in the United States, and it is thought that they might also be applied to heavy commercial vehicles and even to cars. In aircraft, the main problem is the rapid dissipation of the heat generated, as aircraft weighing perhaps hundreds of tons are brought to rest from speeds in excess of 100 m.p.h. Rotors and stators of aircraft disc type brakes have been known to attain temperatures of up to 2,000 deg F in these circumstances. Another factor of major importance in the design of aircraft landing gear is, of course, weight reduction. It is claimed that because the water-cooled brakes are more efficient than the conventional types they can be made significantly smaller and lighter.

Development has been carried out on water-cooled brakes for many years by Roy S. Sandford and Company of Great Hill Road, Oxford, Connecticut, U.S.A. Tests have been carried out on racing and passenger cars, trucks, aeroplanes and industrial machinery. These tests have shown that water-cooling eliminates fade, greatly lengthens brake and lining life and, if desirable, the brake assemblies can be completely shrouded. The new brake is being further developed under licence by the Raybestos Division of Raybestos-Manhattan, Inc., B. F. Goodrich Company, Wagner Electric Corporation and also by the National Supply Company.

The water-cooled drum type brake mechanism is more or less conventional except that in that the lining is bonded to the drum track instead of to the shoe, and the shoe is hollow and has a copper rubbing surface. Hose connections take the coolant into one shoe, then through the other and out to the radiator of the car. The capacity of the radiator does not have to be altered when water-cooled brakes are fitted, and the only change needed in the cooling system is the fitting of an additional inlet and outlet. The brake coolant circuit is boosted by a belt-driven pump, by means of which a circulation of 18 gals/min is obtained through the jackets in the shoes.

A new type of friction material had to be developed for this type of brake, since the requirements are different in many respects from those of conventional brakes. The friction material must be soft enough to give a good frictional coefficient, without causing damage to the copper-lined shoes during operation; at the same time, it must not conduct heat readily so that as high a proportion as possible of the heat generated goes into the shoes and thence into the coolant: this is an important factor so far as the reduction of drum distortion is concerned.

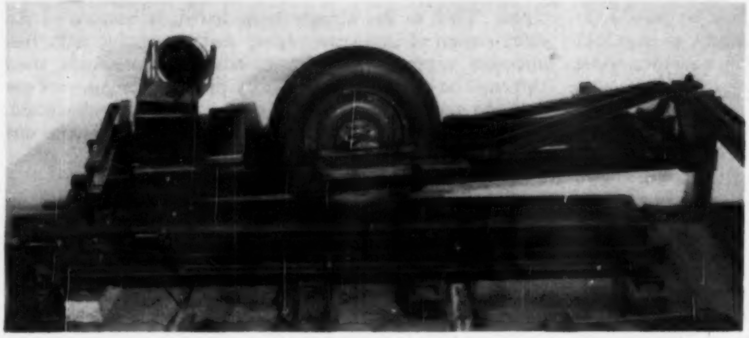
A measure of the success attained in the development of this material is the good results obtained in tests. It is claimed that one set of linings may last well over 100,000 miles. In tests to compare the results obtained with conventional brakes and the water-cooled ones, the following results were obtained. The conventional brake lining wear exceeded 0·140 in. On the other hand, the liquid-cooled brakes exhibited a total wear of only 0·010 in, 0·006 in of which was on the lining and 0·004 in on the shoe: this, it is stated, was mainly the initial bedding-in loss. Subsequently, eleven successive test runs were made and produced negligible wear.

The Wagner Electric Corporation has been developing a system in which the second pump is eliminated. This is done by the employment of a single centrifugal pump to serve both the engine and brake coolant circuits. This manufacturer is also experimenting with coolants that are more efficient than water. These two factors will undoubtedly increase the overall efficiency and simplicity of the system.

The Raybestos Division of Raybestos-Manhattan, Inc. are now applying research to passenger cars. In fact, they have been surveying the market for this brake in kit form. Installation would be simple since the new brakes do not differ greatly from conventional types already fitted. Slots would have to be cut in the back-plate to accommodate the coolant hoses and to allow them to move with the shoes.

A new liquid-cooled disc type brake is also being developed. It is cooled by the fluid that is used to apply the brakes. The system is operated at a pressure of about 150 lb/in^2, and this is supplied by a positive displacement pump. When the brake is not in use, the water is by-passed back to the engine coolant system. Depression of the brake pedal closes the by-pass and this directs the fluid into the brake cooling system. For brake actuation, the pressure is regulated by a throttle valve in the return circuit.

Brakes of this type are smaller than the conventional types that they replace and, of course, they can be fully shrouded to keep out foreign matter. The system is arranged so that in the event of pump failure, the brakes can still be applied by extra depression of the pedal. Further information can be obtained from The Raybestos Division of Raybestos-Manhattan, Inc., Bridgeport, Connecticut, U.S.A.



This illustration is of the test rig for measuring cornering force. Some details of the construction of the rig are shown in the sketch at the foot of this page

LATERAL FORCES ON TYRES

*Distribution of Side-Force and Side-Slip in the Contact Patch of a Pneumatic Tyre**

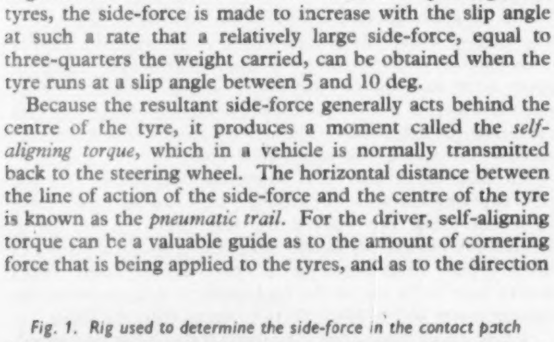
D. H. Cooper, B.Sc.(Eng.), H.Dip.Ed.

In the past, the side-forces on tyres have been studied and discussed in such considerable detail that it should suffice here merely to restate their practical significance. Side-force is useful insofar as it enables a vehicle to be kept on a particular course: to do this, it may be necessary to resist side-wind and road camber. The side-force may, in some instances, be modified by the effect of road wheel camber.

In the first place a side-force is produced by rolling a tyre at a slip angle, that is, so that the plane of the tyre is at an angle to its direction of motion. With most passenger car tyres, the side-force is made to increase with the slip angle at such a rate that a relatively large side-force, equal to three-quarters the weight carried, can be obtained when the tyre runs at a slip angle between 5 and 10 deg.

Because the resultant side-force generally acts behind the centre of the tyre, it produces a moment called the *self-aligning torque*, which in a vehicle is normally transmitted back to the steering wheel. The horizontal distance between the line of action of the side-force and the centre of the tyre is known as the *pneumatic trail*. For the driver, self-aligning torque can be a valuable guide as to the amount of cornering force that is being applied to the tyres, and as to the direction in which the wheels are steered from the neutral position.[3]

Fig. 1. Rig used to determine the side-force in the contact patch
A *plane of wheel;* **B** *slip angle;* **C** *interrupter to record progress;* **D** *single leaf spring*



Whereas a number of authors have described how side-force and self-aligning torque vary with slip angle, vertical load, inflation pressure, size of tyre and rim, as well as with the condition of the road surface, this paper describes the front-to-rear distribution, within the contact patch, of the cornering force. The intensity of cornering force, in turn, determines the magnitude of total cornering force, self-aligning torque and trail. Work carried out by The Dunlop Rubber Co. Ltd. on cornering force was published first in 1949,[1,2] and more extensively during and after 1954.[3–8] This work can now be correlated with that of H. Martin,[9] H. Fromm,[19] B. V. Schlippe,[20] and P. Kraft,[10] who, between 1935 and 1941, published diagrams of the distribution of horizontal forces and movements in the contact patch of a tyre rolling along a straight path and at a slip angle.

Since that date, however, Dunlop have demonstrated[5] that, at small slip angles, even at ½ deg, the forces due to inward and outward movement, or shuffle, of the tyre tread are unimportant by comparison with those due to cornering, Fig. 8. The relative insignificance of slip due to shuffle during rolling can also be seen from the absolute values of 0·2 cm or so, obtained by Kern.[18]

Chiesa[11] has pointed out the large part that cornering forces play on road vehicles during normal journeys, and V. E. Gough[4,5] has discussed the overridingly important role played by cornering forces in causing tread wear. A dissertation on friction has been given by Gough.[17]

Theoretical background

A theoretical study of the development of side-force and camber thrust has been published by Fiala,[12] and although we agree with much of what he states, we do not agree with the assumption that

$$\frac{\text{shear stress in contact patch}}{\text{vertical pressure in contact patch}} = \text{constant.}$$

In fact, at the Dunlop Research Centre, we have established what we believe to be a more precise relationship between shear stress and vertical pressure. Nor do we agree with

**Translated from* Kautschuk und Gummi, *October*, 1958.

the assumption that the front portion of the contact patch is free from side-slip.

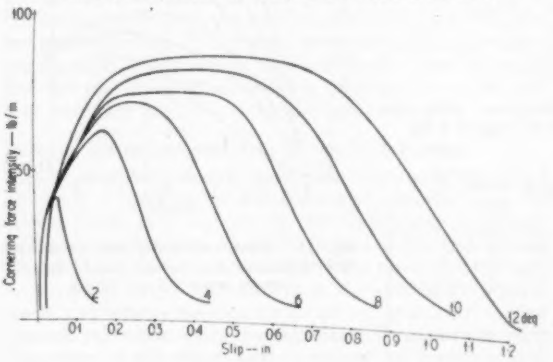
A paper on tyre testing, by A. G. Fonda,[13] should also be mentioned. Although it contributes extensive data, it does not offer much explanation of the development of side-force. More pertinent, perhaps, to our paper is Fonda's analysis of the lateral deflections of the tread in the contact patch, based on the theory of the taut string. Although it is not proposed to enlarge on this at present, we do not believe that the taut string hypothesis is the best way to simulate the conditions in a tyre, and we hope to offer an alternative theory on another occasion.

Cornering force rig

Since a description of the cornering force rig has already been published,[8] only a brief account is necessary here. The rig consists of a plank about $2\frac{1}{2}$ m long and 28·7 cm wide. This plank is mounted on rollers, so that it is free to run for a distance of 1·6 m at 3 cm/sec, Fig. 1. Above the plank is a framework, which is hinged at one end; in this framework is carried an assembly comprising an axle, wheel and tyre. The tyre is free to roll on the plank and can be held, at any slip angle up to 12 deg, by an adjustable stop at the free end of the frame.

Near the end of the plank, the tyre rolls over a bar, the emery-covered top surface of which is precisely level with the emery-covered top surface of the plank. This bar is mounted on leaf springs, in such a manner that it can deflect a small amount laterally under the influence of the cornering force, but the springs cannot deflect vertically. The lateral motion is magnified and transmitted by levers to the Y axis of a graph, which is drawn on a glass slide.

Fig. 3. Reproduction of a diagram of cornering force intensity plotted against side-slip, at slip angles up to 12 deg, for the 6.70–16 tyre inflated to a pressure of 16 lb/in² and loaded to 6·1 cwt

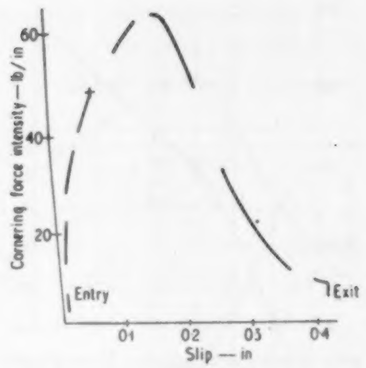


In addition, the tread of the tyre is penetrated by the upstanding prongs of a fork, which is thus moved by an amount equal to the side-slip. In this way, the side-slip of the centre rib of the tyre, as it rolls over the fork, is transmitted to the glass slide, which, by its movement relative to the scribe, records the side-slip from front to rear of the contact patch.

At the same time, a contact breaker at the side of the plank interrupts the tracing of the slide at intervals corresponding to movements of one inch of the plank relative to the tyre axle. Another device registers the position of the wheel centre relative to the contact patch.

Much of the preliminary work on this subject is now complete and experience has been gained from testing some seventy tyres. For example, at an early stage, it was established that the lateral motion of all the ribs in the tread-band of a tyre is nearly equal, when compared with the overall movement due to side-slip; also, if required, the cornering force can be made to rise to its full value before its measurement is begun.

Fig. 2. Reproduction of a diagram of cornering force intensity and side slip, with a 6.70–16 tyre inflated to 16 lb/in² pressure, loaded to 6·1 cwt, and set at an angle of 4 deg



Test Method

When the plank is drawn under the tyre, a graph of cornering force intensity, lb/in, plotted against slide-slip, is drawn on a glass slide, Fig. 2. The curve is interrupted at every inch of forward movement of the plank, and two marks are made on the slide to indicate the position of the wheel axle in relation to the beginning and end of the contact patch. A family of such curves for slip angles of 2 to 12 deg is shown in Fig. 3. Each curve is the average of four results taken at four positions round the tyre; this reduces any discrepancies due to lack of uniformity of the tyre.

Cornering force intensity at various slip angles

From the original curve, it is possible to derive a curve, Fig. 4, of cornering force intensity plotted against the distance X measured backwards and forwards, from the centre-line of the wheel. The rear half of such a curve is an approximate indication of the length of the contact patch. In this example, the length of the contact patch is $3{\cdot}7 \times 2 - 1 = 6{\cdot}4$ in. It is necessary to subtract one inch because the bar over which the tyre rolls is 1 in wide. The length of contact patch, measured from a contact print, is 6·6 in.

At 2 deg slip angle, the cornering force intensity distribution is approximately triangular in form. It develops slowly and rises to a peak, which is behind the centre-line of the wheel. At 4 deg slip angle, Fig. 5, the force is larger, but its distribution diagram is still triangular. Then, at 6 deg and 8 deg, the peak of the triangle is less sharp. At 10 deg, and particularly at 12 deg, there is a distinct flat on the top of the diagram, which therefore is no longer a triangle but has become a trapezium. The appearance of the flat signifies that the cornering force intensity can increase no further, because the frictional force has reached its limiting value. Loss of cornering force intensity at the end of the contact patch is, of course, the result of the reduced vertical pressure in that region, as the tread-band leaves the ground.

Cornering force, self-aligning torque and trail

The area under the cornering force intensity curve is the total cornering force, Fig. 4, and it is evident that at most times, this force acts behind the centre of the wheel. In fact, the line of action of the force is the line *ab* through the centre *G* of the area under the curve, and the distance *t* is the pneumatic trail. The self-aligning torque is the product of cornering force and trail.

Slip angle and cornering properties

The effect of the increasing slip angle on cornering force, trail and self-aligning torque has been determined many times in the past by running a tyre on a drum.[1–4,7,8,12,13,14]

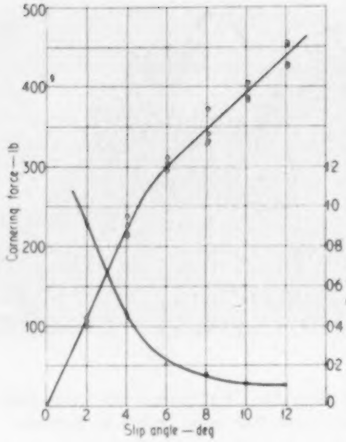
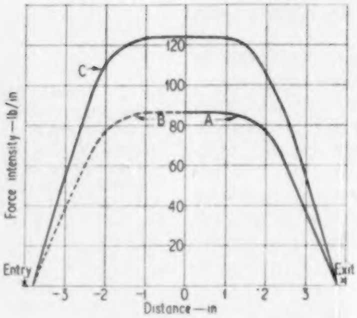


Fig. 6. Curves illustrating cornering force and trail at slip angles up to 12 deg



All the curves on this page are test results obtained with a 6.70–16 tyre inflated to a pressure of 16 lb/in² and loaded to 6·1 cwt

Fig. 8. Diagram of the distribution of the vertical force over the tyre contact patch

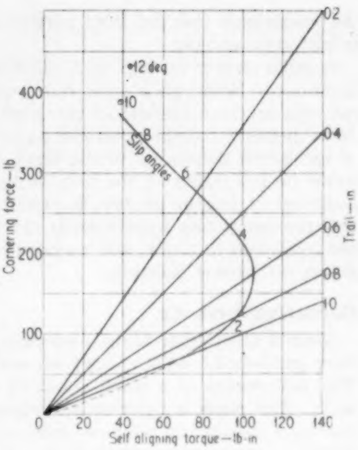


Fig. 7. This diagram shows cornering force plotted against self-aligning torque

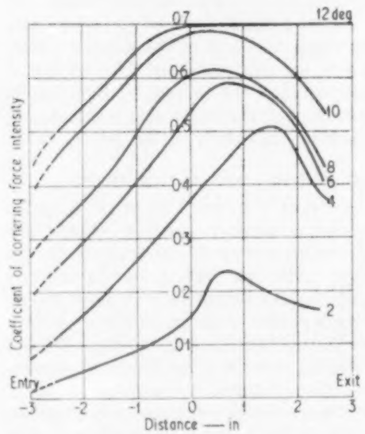


Fig. 9. Coefficient of the cornering force intensity along the contact patch, for slip angles ranging from 2 to 12 deg

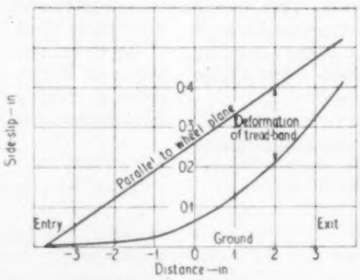


Fig. 10. Side-slip distribution along the contact patch for a slip angle of 4 deg

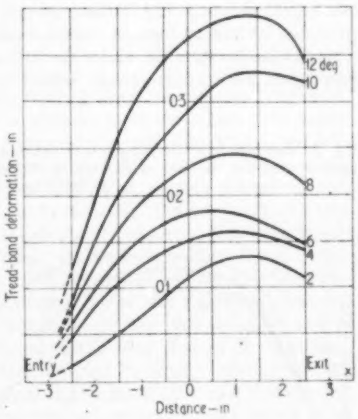
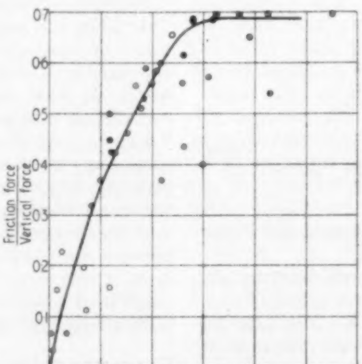
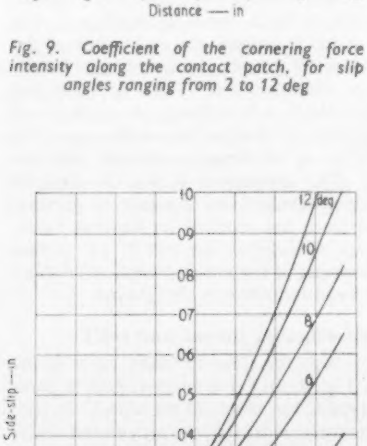


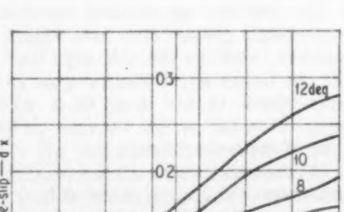
Fig. 11. Curves indicating the tread band deformation in the contact patch, for slip angles between 2 and 12 deg



Above: Fig. 14. Coefficient of cornering force intensity plotted against ds/dx for the 2 to 12 deg range of slip angles



Left: Fig. 12. Distribution of side-slip at slip angles between 2 and 12 deg



Right: Fig. 13. Distribution of the rate of side-slip for slip angles of 2 to 12 deg

When the tyre rolls on a flat surface, the properties are, broadly speaking, the same as on a drum.[15] The gradually-changing shape of the cornering force accounts for the slope of the curve of total cornering force plotted against slip angle, as well as the reduction of trail with slip angle, Fig. 6.

Plotting curves of cornering force against self-aligning torque, Fig. 7, published first in Great Britain[3] and subsequently in Germany,[4,14] is a convenient way of relating these quantities, together with pneumatic trail, on one graph.

Vertical force distribution

The distribution of vertical force over the contact patch has been studied relatively little—once by Martin,[9] who drew the distribution of force along two fore-and-aft planes, once by Kraft,[10] and once by Teller and Buchanan.[16]

It is assumed that when the cornering force intensity for 12 deg slip angle reaches the flat on the curve, Fig. 5, the coefficient of friction is at its maximum value. It is also assumed that in the rear half of the contact patch, A in Fig. 8, the coefficient remains at or near the maximum value and that the vertical force distribution diagram is the same shape as that of the horizontal force distribution.

On the assumption that the pressure distribution is symmetrical, the mirror-image B of the horizontal force distribution, Fig. 8, has been drawn first, and then a second curve, C in Fig. 8, has been drawn, with increased Y-ordinates, so that the area under the curve is equivalent to the vertical load on the tyre.

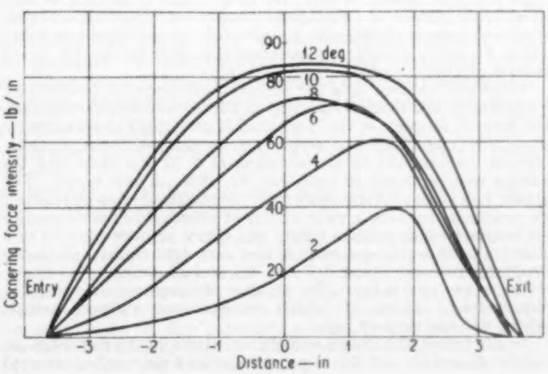
Cornering force intensity coefficient

Let M be the cornering force coefficient, which is defined as the ratio of cornering force intensity, at any point along the contact patch, to the vertical force intensity at the same point. Then, the values of M at each point from front to rear of the contact patch, over the wide range of slip angles available, are plotted in Fig. 9. It should be noted that the M values are not high. They would have been higher were it not for the fact that in the particular test illustrated here, the tyre was purposely under-inflated and under-loaded.

Lateral distortion and slip of the tread-band

It is possible to derive, from each of the original curves in Fig. 2, a curve of the distribution of side-slip along the contact patch, as in Fig. 10. In this latter graph, the X axis indicates the direction of relative motion of the ground, and in addition a line is drawn parallel to the plane of the wheel. At 4 deg slip angle, the tread, from the start of the contact patch, first follows the road for about 4 in and is then pulled towards the wheel plane by the elastic forces in the tyre. Distortion of the tread-band is worthy of further study, Fig. 11. A family of curves of the distortion of this tread while it is in the contact patch is given for slip angles up to 12 deg, Fig. 12.

Fig. 5. The cornering force intensity, at slip angles up to 12 deg, for the 6.70–16 tyre inflated to 16 lb/in² and loaded to 6·1 cwt



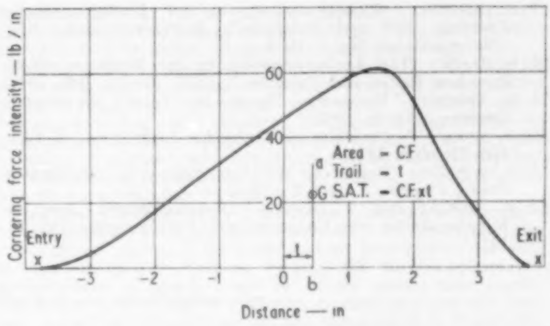


Fig. 4. Curve showing distribution of cornering force intensity along the contact patch of a 6.70–16 tyre inflated to 16 lb/in² pressure, loaded to 6·1 cwt and set to run at a slip angle of 4 deg

From the family of curves showing the distribution of the side-slip along the contact patch, Fig. 12, it is interesting to note, first, that slip increases from front to rear and, secondly, that, at slip angles above 8 deg, the whole of the tread rubber is, in fact, sliding sideways over the ground.

If the wheel is moving forward at a uniform velocity V in/sec, the distance X in, measured along the contact patch, represents also the time X/V sec. The slope of the slip curves is proportional to the rate of sideways slip divided by the rate of forward rolling motion:

$$\frac{ds}{dx}=\frac{ds}{dt}\times\frac{dt}{dx}$$

$$=\frac{1}{V}\frac{ds}{dt}$$

$$=\frac{\text{rate of side-slip}}{\text{rate of forward motion of the axle}}$$

Curves of rate of slip with respect to distance moved through the contact patch are obtained by differentiating the slip curves and plotting ds/dx against x, as in Fig. 13.

Finally, in Fig. 14, cornering force intensity coefficient has been plotted against the rate of side-slip divided by the rate of forward rolling and, allowing for experimental errors, a single curve has been obtained. Since cornering force intensity coefficient is:

$$\frac{\text{frictional force at any point}}{\text{vertical force at any point}}$$

the curve shows how the frictional force is related to side-slip.

From this curve, it can be seen that the frictional force at any point in the constant patch of a tyre rolling at a steady speed is not constant but increases, from zero to a maximum, with the rate of slip. After reaching its maximum value, it remains constant, but at vehicle speeds over 45 m.p.h., it begins to fall slightly.

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Recent German Publications

Brief Reviews of Current Technical Books

Entwicklungsprobleme der Kraftfahrzeugmotoren. (Development Problems of Vehicle Engines.)

In German. By Alfred Jante.

BERLIN: AKADEMIE-VERLAG, Mohrenstrasse 39, 1958. 11¾ × 8¼. 38 pp. D.M. 6.80.

This is a paper that has been read before the Mathematical, Physical and Technical Section of the German Academy of Science. The author is Professor at and Director of the Institute for Internal Combustion Engines and Vehicle Engineering at the Technical High School, Dresden. Two previous publications by this distinguished engineer, teacher and scientist were reviewed in the February 1956 and April 1957 issues of *Automobile Engineer*. In these reviews, attention was drawn to one of the author's major preoccupations, the possibilities of reducing the part-load fuel consumption of road vehicles. The paper now under review is divided into three major sections; one is devoted to the engine characteristics, the second to fuel consumption and the last to turbo-engines. In a short concluding note, the economics of vehicle engines is considered.

In the first section, the author gives a strictly logical evaluation of the engine torque characteristics in terms of vehicle requirements, with particular reference to acceleration and its evaluation. In recent years, so much sense and nonsense has been written on the desirable shape of engine torque curves, including fine differentiations to the effect that what is of importance to tractors is of less importance to cars and of no importance to rail vehicles, that the author's clear and rigorous analysis of this very important aspect of engine performance should clarify the issue. Thus it undoubtedly is of considerable benefit to designers and users alike in that it fosters clear thinking unburdened by intuitional considerations. The question of engine flexibility is, of course, closely bound up with the shape of the torque curve; here, too, different interpretations of flexibility have led to different conclusions. The author endeavours to clarify this aspect of engine performance, and in addition indicates ways of improving the flexibility, thus simplifying the task of transmission designers.

The next section deals in the main with the important but rather neglected subject of fuel consumption under part-load conditions, a subject that is becoming increasingly important owing to the prevailing trend towards the use of engines of higher output. Here the author considers the possibility of adjusting the number of working cylinders to meet the load requirements, and thus obtaining reduced fuel consumption; he quotes test results obtained with a six-cylinder engine running on five, four and three cylinders. It might be mentioned that this method of reducing part-load fuel consumption was also considered by the late E. A. Chudakov in 1947; recent tests carried out at Cheliabinsk by Smolkin, Astahov, Danilievich and Gonenko suggest that this method, while capable of improving the engine economy by as much as 20 per cent, requires complex design alterations. The author also considers the beneficial effects of supercharging, the use of a heat exchanger between exhaust and inlet, alterations to the combustion chamber of an existing engine with already good fuel consumption characteristics, and torch ignition as investigated by N.A.M.I. in Moscow. These considerations are followed by an examination of the advantages to be gained by increasing the compression ratios of petrol engines up to 16 : 1 and a comparison of the fuel consumption characteristics of petrol and compression ignition engines. The section concludes with an examination of the relationship between optimum fuel consumption limits and transmission gear ratios.

Turbo-engines, comprising pure gas-turbines and free-piston generator-turbine combinations, are briefly considered, again with particular reference to part-load fuel consumption, and it is concluded that so far as vehicles are concerned, if economic operation is to be the aim in this field of application, the gas turbine can only be used in conjunction with piston engines. Since a similar flexibility can be attained by the use of reciprocating engines in conjunction with hydraulic transmissions, the characteristics of this combination are also considered.

The paper concludes with the reminder that, during the current year, the total number of the world's road vehicles has comprised some 79 million passenger cars and some 24 million lorries and buses, with a total power output of about 4 million mega-watts. Since this is about seven to ten times as much as the total output of the world's power stations the importance of keeping to a minimum the fuel consumption of this fleet scarcely needs stressing. The author is to be congratulated on the production of a well-balanced, clear, concise and intensely thought-provoking paper, the lucidity of presentation and originality of outlook being particularly impressive.

Kräfte in Errichtungen zur Verbindung von Kraftfahrzeugen mit Mehrachsanhängern über 20 km/hr Höchstgeschwindigkeit. (Forces in Couplings between Motorcars and Trailers at Speeds over 20 km/hr.)
Deutsche Kraftfahrtforschung und Strassenverkehrstechnik No. 113.

In German. By Otto Bode, Prof. Dr. Ing., and Heinrich Meyer, Dipl. Ing.

DÜSSELDORF: VDI-VERLAG G.m.b.H., 1958. 11¾ × 8¼. 37 pp. Price D.M. 15.40.

The forces encountered in the couplings between motor vehicles and trailers affect the reliability of the train under driving and braking conditions. In addition, with buses hauling trailers, these forces are also of importance with regard to steering as well as to the comfort of passengers. Because of this, the German road regulations, *Richtlinien für die Prufung von Fahrzeugteilen, April* 27, 1957, specify requirements that have led to the use of certain officially licensed coupling designs.

To obtain reliable data for the dynamic stressing of coupling components, the Federal Ministry of Transport asked the Institute for Road Transport, of the Technical High School at Hanover, to carry out tests. The first series of tests dealt with the dynamic stresses encountered in the coupling of a 32 ton train and the results were reported in issue No. 92, 1956, of this series. The report now under review deals with the results of tests carried out on couplings between a wide variety of vehicles. These comprise six lorries weighing, when empty and laden, respectively, 2·6 to 8 and 5·3 to 16 tonne and two tractors and eight trailers, the latter weighing between 1 and 5·7 and 4·3 and 24 tonne respectively, when empty and laden. The number of couplings amounted to twenty-four, twelve of which incorporated rubber springs, whilst five had ring-springs.

In the report, the design and characteristics of the couplings are briefly discussed, and the test procedure and the method adopted

for test evaluation are dealt with. The main body of the report deals with the longitudinal forces in the couplings. These are evaluated individually for each vehicle type, and the principal results are plotted, in terms of maximum force, against trailer weight for both good and bad roads. The forces encountered as the result of full brake application, lateral forces due to braking as well as those encountered during fast runs, and the vertical forces encountered on the road and during braking, are also considered in some detail. The report concludes with an overall evaluation of the results on a statistical basis.

An interesting outcome of the investigation is the finding that when the vehicle is running over poor roads the stresses in the couplings can be up to four times greater than on well-maintained roads, that the forces are proportional to the speeds, that the load, and particularly, the load on the motor vehicle, can increase the forces two-fold. It is of interest to note that, as a first approximation, the coupling forces encountered at about 25 to 30 m.p.h. on a good road are equivalent to those on a poor road at about 12 m.p.h. So far as lateral forces are concerned, attention is drawn to the possibility of reducing their magnitude by the use of couplings with suitable lateral control springs.

The quantity of actual test results analysed and presented in a logical and concise manner will make this report a welcome source of information to vehicle chassis and component designers, who will be grateful to the authors for the trouble they have taken in planning and carrying out the tests and in the presentation of the results.

Kräfte in Kugelgelenk—Flächenkupplungen bei Omnibuszügen mit Zweiachsanhängern. (Forces in Ball-and-Socket Flat Face Couplings between Buses and Four-wheel Trailers.) Deutsche Kraftfahrtforschung und Strassenverkehrstechnik, No. 114.

In German. By Otto Bode, Prof. Dr. Ing., and Heinrich Meyer, Dipl. Ing.

DÜSSELDORF: VDI-VERLAG G.m.b.H. 1958. 11¾ × 8¼. 20 pp. Price D.M. 11.20.

Following Part I, Report No. 92, and Part II, No. 113, this report deals with the results of tests carried out, by the Institute for Road Transport at the Technical High School at Hanover, at the request of the Federal Ministry of Transport. Its object is to specify the stresses encountered in couplings of passenger carrying road trains, as used during rush hours in some central European city and suburban services. Apart from the fact that reliable data is obviously valuable, the subject became urgent owing to coupling and drawbar breakages encountered with bus trailers of a certain design.

This report deals with the test procedure adopted when determining the forces encountered in the couplings of three different bus trailers. All three were fitted with ring-spring drawbars, coupled to a Büssing vehicle and operated over good and poor roads, as well as on an Autobahn section.

Equipment used and the methods adopted for the evaluation of the test results are discussed. The results obtained are considered in detail, with particular reference to the lateral forces encountered by the drawbar of the trailer coupling, with which fractures were experienced in some instances after only 12,000 miles. It is shown, on the basis of calculations as well as the evaluation of test results, that the coupling arrangement of the particular trailer is subjected to lateral forces of a magnitude such as to cause fatigue failure. The subject matter of the report is a valuable addition to that in the previous two reports dealing with the same subject, particularly since it also considers a number of design features relating to trailer steering, and their effect on the forces encountered by the couplings.

Untersuchungen an Mechanischen Lenkungen und Hilfskraftlenkungen Schwerer Kraftfahrzeuge. (Investigations of Mechanical Steering Systems and Auxiliary Power Steering Systems of Heavy Vehicles.) Deutsche Kraftfahrtforschung und Strassenverkehrstechnik. No. 115.

In German. By Otto Bode, Prof. Dr. Ing. and Gerhard Bode, Dipl. Ing.

DÜSSELDORF: VDI-VERLAG G.m.b.H., 1958. 11¾ × 8¼. 32 pp. Price D.M. 14.90.

As a result of the increasing traffic density and the stress imposed on drivers of lorries and public service vehicles, it has become necessary to determine reliably the forces that can be considered as acceptable so far as the steering effort is concerned, and *inter alia* the necessity or otherwise of fitting auxiliary power steering. To clarify this important subject, extensive road tests were carried out, at the request of the Federal Ministry of Transport, by the Institute for Road Transport of the Technical High School at Hanover. These tests were aided by the industry and by a number of city transport undertakings. The results merit a careful study by vehicle builders and operators.

After considering the general requirements that must be met by the steering gear, to ensure satisfactory operation so far as the driver is concerned, the authors consider the effect, on driver fatigue, of driving heavy vehicles. The method adopted for the measurement of driver fatigue consisted in measuring the pulse frequency throughout the test; this was done by fixing a lamp and photocell to the driver's ear lobe, the opacity of which alters with the blood impulses through it.

Equipment devised to record the torque at and the displacement of the steering wheel are also described. To investigate the driver fatigue aspects of the problem, tests were carried out with an auxiliary power steering system, the force-displacement characteristics of which could be altered within wide limits. The bus concerned had an automatic transmission and the tests were carried out over a 6 mile city run requiring 20 to 23 minutes for its completion. Of the six drivers tested, four were of average physique, while one was of slight and the other of heavy build. The results of the tests were analysed and the personnel concerned questioned; in particular, some 50 drivers were questioned regarding forces and action of steering wheels. It was concluded that to ensure satisfactory operation, the steering effort should be proportional to the steering angle of the wheels, the maximum effort, at maximum angle, should not exceed about 25 to 30 lb, while the number of turns from lock-to-lock should be limited to five or six. However, owing to the fact that the test runs were of a particularly difficult nature, and also owing to economical considerations, it was concluded that the maximum permissible force can be increased to 55 lb; but it will be desirable to reduce this as technical development proceeds.

The next main section of the report deals with the requirements to be imposed, and the principal data of the five vehicles tested and of the various power systems used; these include the Ate-Ross, Fulmina, Ate-Hydross, ZF-Spindle hydraulic and ZF-Gemmer systems. The results are plotted in terms of steering angle, against force at steering wheel, for various speeds and front wheel loads. Opportunity was also taken to measure the bending moments in the steering gear. Comparative tests were made with and without power assistance, and the tests were extended to include a road circuit incorporating numerous curves. The plots and legends should be clear even to readers not familiar with German. In conclusion, suggestions are made for a method of testing the effectiveness of steering gears. Altogether this is a timely and thorough publication on an important subject and can be highly recommended to designers and research engineers.

Vom Motor zum Auto, Fünf Männer und ihr Werk. (From Engine to the Car, Five Men and their Work.)

In German. By Eugen Diesel and others.

STUTTGART: DEUTSCHE VERLAGSANSTALT. 1957. 337 pp. D.M. 8.

This book deals with the five great German designer-engineers whose names have become household words wherever motor cars, aviation motors and combustion engines are studied. Their names are: N. A. Otto, G. Daimler, K. Benz, R. Diesel and R. Bosch. In the book, their lives and studies are discussed, without the use of too much technical jargon, by three experts including Eugen Diesel, who has used in condensed form material from his older work entitled "Die Geschichte des Diesel Personenwagens." The book succeeds admirably in conveying, partly by skilfully prepared illustrations, the brilliance of the inventions, the gradual progress made, and the logic of the improvements as they follow one another. It also expresses the loneliness of the original inventor mind, forging ahead whatever the reactions of a sceptical world, and it deals with the social impact of industrial exploitation of the inventor's ideas. Not the least merit of this biographical work is that it is never too technical nor does it permit the reader to forget the future perspectives of our motorized existence. The work was commissioned by the Friends of the Technische Hochschule, Stuttgart, with a subsidy from the Max Kade Foundation, New York.

Die Geschichte des Diesel Personenwagens. (The History of the Diesel Private Car.)

In German. By Eugen Diesel.

STUTTGART: DEUTSCHE VERLAGSANSTALT. 1955. 123 pp. Price D.M. 8.40.

Rudolph Diesel's quest for and the development of a high-speed automobile engine is the subject of this book, which is a biography. His failure to achieve his aim and the signal improvements made by Daimler and Benz are also discussed. Although the work is written by a son about his father, it is far from adulatory in tone; in fact it is remarkably objective and detached.

SIMPLIFIED SEALING

Ingenious Applications of Dowty Synthetic Rubber Rings by the Ford Motor Co. Ltd.

EVEN a relatively simple internal combustion engine incorporates an exceedingly complex plumbing system. Within the confines of a space of small volume, quantities of air, fuel, lubricating oil, water, combustible fuel-air mixture, and exhaust gases are, severally, in continuous or intermittent flow or are in continuous circulation. To keep these media separately intact, under conditions of widely varying pressures, temperatures, and flow velocities in an assembly of component parts subjected to rapidly fluctuating mechanical stresses, presents difficult problems of design.

Quite effective but relatively complicated seals developed in the past for various applications have given satisfactory service. However, to meet the requirements of modern high-rate production, with its insistence on economy and on ease of assembly, the possibility of simpler but no less effective devices is being carefully investigated. An example of the progress currently being made is provided by the Ford Motor Co. Ltd. Their engineers were quick to take advantage of the development of flexible sealing materials, particularly of the synthetic rubber type, in which great progress has been made in recent years. The versatility of these products is such that, with careful selection, they have not only replaced much more complicated mechanical seals without loss of efficiency, but, in some cases, have simplified assembly and access problems by eliminating more permanent types of joint.

A typical case concerns the oil pipe feeding the overhead valve gear on the current Consul, Zephyr and Zodiac engines. This pipe, illustrated at the right of Fig. 1, supplies oil from a gallery on the off-side of the engine. For ease of servicing, the pipe must be removable with the cylinder head and, in fact, is installed during assembly already attached to the valve gear. The lower end of the pipe, however, must be securely sealed in the oil delivery hole in the engine block, where it is subjected to high oil temperature and pressure, and to vibration. This is accomplished by the use of a simple O-ring of synthetic rubber, produced by Dowty Seals Ltd. of Ashchurch, Gloucester. The ring, three examples of which can be seen alongside the dismantled pipe, is simply retained between two circumferential ridges swaged in the copper tube itself, and is easily slipped on by the fingers. When the lower end of the pipe is inserted into the oil hole, the ring is gripped between the tube and the walls of the hole to provide a resilient seal.

The properties of the ring material, basically an oil-resistant synthetic polymer, are modified during formulation and curing to produce a comparatively firm seal, which under slight compression mates perfectly with the surrounding metal. Natural rubber, although suitable from the mechanical point of view, would soon decompose when exposed to oil and vapour fumes in the engine. Although synthetic rubbers have always shown promising resistance to petroleum products, much progress has now been made in developing formulations which combine oil-resistance with the resilience of the natural material.

This example is of a static seal, which is subject to no mechanical movement apart from vibration and the slight adjustments due to thermal expansion and contraction. O-rings of similar type can be designed to be equally effective for dynamic seals, for which purpose they are very much simpler than the traditional types. O-rings are also better able to accommodate fairly wide dimensional tolerances, thus permitting further production economies.

An excellent example is shown in Fig. 2. Here the distributor, on the same types of engine, is about to be inserted in its mounting, the lower end of the distributor shaft engaging with the usual socket in the drive spindle. To prevent oil escaping from the crankcase an oil seal is required between the outer sleeve and the mounting bore. Effective sealing is provided by the O-ring, which can be seen near the top of the outer sleeve, where it is located in a circumferential groove.

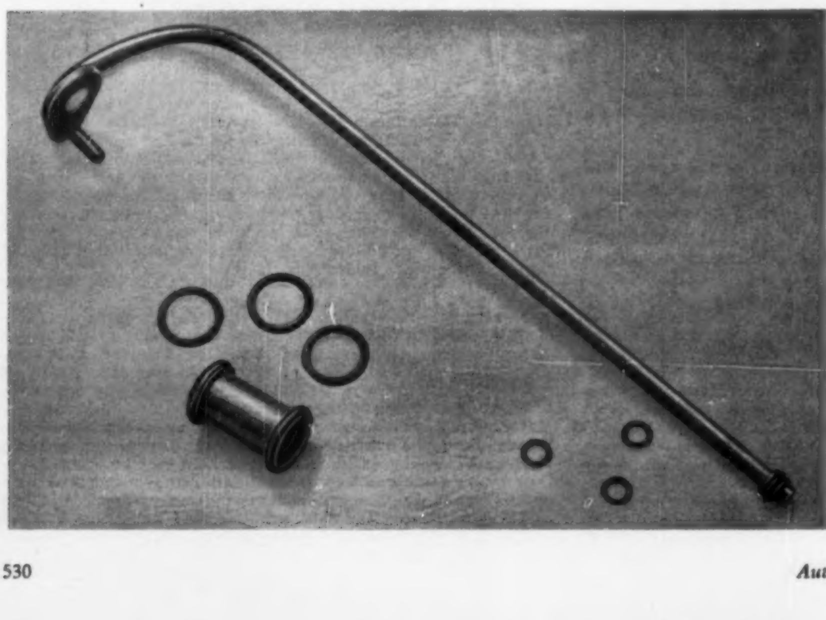


Fig. 1. The curved pipe (right) conveys oil to the valve gear on Ford engines. The short tube (left) forms a by-pass conduit for cooling water. Both are sealed in position by the Dowty synthetic rubber O-rings shown

Accuracy and stability of dimensions are essential in an O-ring to ensure efficient sealing. In fact, the dimensional tolerance on rings of this type amounts to only 0·005 in, while the flash residues from the moulding process, lying along the inner and outer circumferences, never exceed 0·005 in wide and 0·003 in high. The surface finish overall is very smooth. Wear on the ring is negligible, and it has proved fully capable of lasting the overhaul life of the engine.

A different function is performed by the seals illustrated in the lower part of Fig. 1. The short length of pipe forms the by-pass conduit for coolant when the thermostatic valve closes the riser to the radiator. O-rings of synthetic rubber, located by swages in the tube, form water seals for the pipe which is firmly clamped in place when the thermostat housing is bolted down. It will be obvious that considerable expansion and contraction take place at this point on starting from cold and on cooling down; the O-rings accommodate the resulting movement between pipe and casting. It will also be noted that economy is here served, without loss of efficiency, by the elimination of a separate fixing operation, while service work is simplified by the ease of dismantling and re-assembly.

Another O-ring which fulfils both sealing and securing functions is found on the oil breather pipe on top of the timing case at the front of both the diesel and petrol engines used in Thames Trader trucks. In Fig. 3, the ring, which must be a tight fit on the pipe, is being pushed into place by means of a tapered mandrel on which several rings can be accommodated at one time for rapid assembly on successive engines. When the ring is in position, a crankcase breather air cleaner is slipped over the top of the pipe, as shown in Fig. 4, the ring itself being the only holding device used. The grip provided by the seal holds the fitment firmly in place despite the severe, large-amplitude vibration experienced during starting and idling, as well as the higher-frequency vibration at speed. The sealing function is to prevent oil mist from draining down the outside of the pipe, and thus to ensure that all oil is returned to the timing cover.

Another Dowty seal on the Trader truck range of vehicles consists of a special-section ring designed to prevent the escape of grease from and the ingress of dirt to the steering king pin bearings. A conventional design of steering swivel is employed, the king pin being anchored in the eye of the axle beam by a cotter bolt. The seal, at the top of the pin consists of a square-section, synthetic rubber ring having a small, annular groove in the lower face. It is held in place by a dished steel washer above it and both ring and washer are retained by a stiff wire circlip.

The king pin is driven down through the spindle body and axle beam eye, as shown in Fig. 5, using a soft-headed hammer, until the dished washer so compresses the sealing ring that it spreads to fill the gap between washer and

Fig. 2. Assembling the ignition distributor. Near the top of the spindle sleeve is an O-ring which prevents the escape of oil from the crankcase

Fig. 3. Mounting an O-ring on the breather pipe from the timing cover of a Thames Trader engine. The ring, a tight fit, is expanded on a tapered mandrel and 'rolled' into its recess

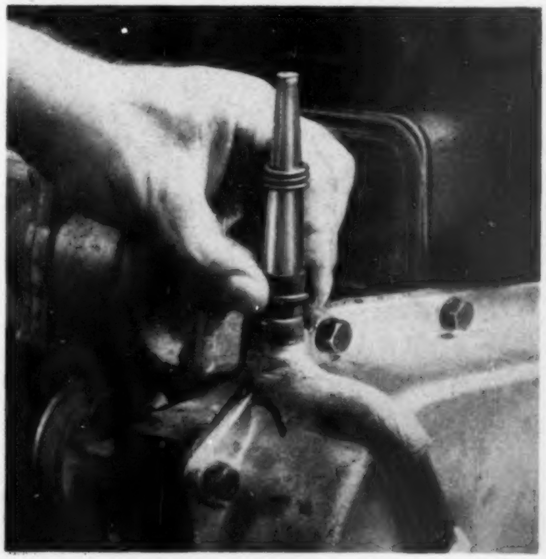
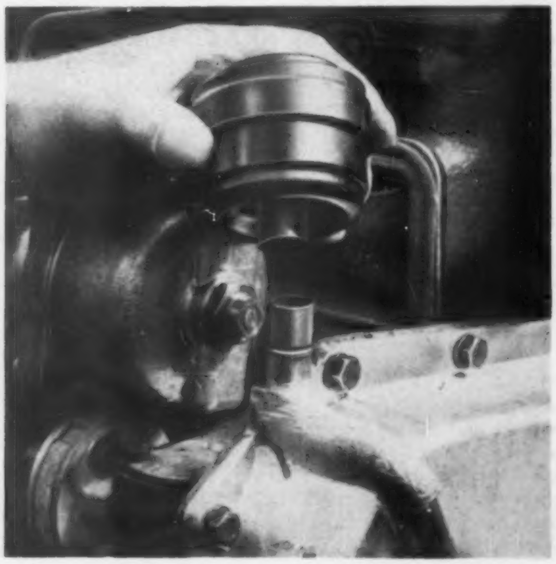


Fig. 4. The ring in position on the breather pipe and the cleaner about to be slipped over the top. The friction of the O-ring holds the cleaner securely in place in service



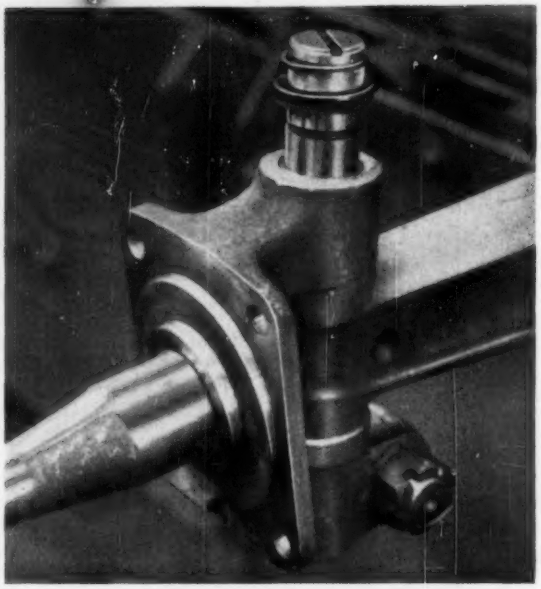


Fig. 5. Insertion of the king pin. When driven home, the circlip forces the dished washer over the ring, which deforms to seal the top of the bush

spindle body over the full width of the washer. The sealing ring section is so designed that, under pressure from the grease gun, lubricant may just issue from the top of the bush, so showing that the bearing is fully lubricated, while grit and water cannot enter in service. The bottom end of the king pin bush is sealed by a welch plug forced into a recess in the lower end of the spindle body.

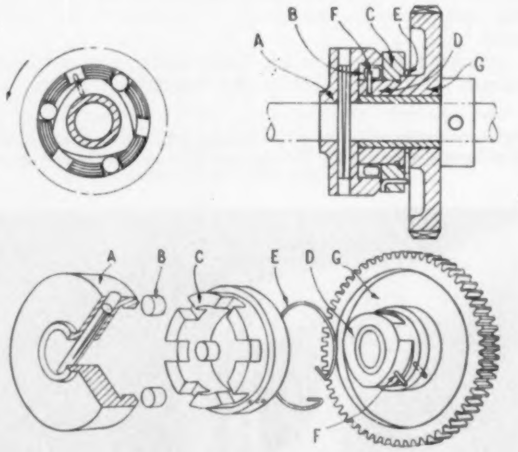
In the case of a heavily laden vehicle this seal has an important function to perform since good lubrication is essential to keep down the steering effort required, apart from the question of reducing wear to the minimum. The sealing ring is able to withstand the attack of greases almost indefinitely under conditions of permanent distortion, wide variations in temperature and the presence of moisture, sometimes heavily salt-laden as a result of snow clearing operations.

All the seals described are comparatively simple in design. They are easier to accommodate and to apply than more intricate, composite seals. This apparent simplicity, however, was made possible only by intensive development of the basic materials. The firm responsible has long experience in the design of sealing devices to very stringent requirements for aircraft applications and is now able to pass on the benefits of long-term research work in that field to the automobile industry. The scope of flexible seals is constantly widening and a very extensive range of standard sizes of O-rings in natural, synthetic, and silicone rubbers is available. Square-section and U-shaped rings, gaskets and other special seals can be designed to cover almost any requirement involving the retention of fluids or gases at the highest pressures met with in engineering. They offer, it would seem, opportunity to save cost, space, assembly time, and, possibly, a reduction of maintenance in service.

FREE WHEEL FOR FRACTIONAL HORSEPOWER APPLICATIONS

A FREE-WHEEL clutch, capable of handling ½ h.p. at 1,500 r.p.m., has been designed and is manufactured by Tiltman Langley Ltd., of Redhill Aerodrome, Surrey. This free-wheel is said to be capable of operating satisfactorily at temperatures as low as – 40 deg C, as well as under tropical conditions.

The essential components of the assembly are shown in the accompanying illustrations. A housing A forms the outer component of the free-wheel. It is pinned to the shaft and contains the rollers B, their cage C and the cam D. Normally the spring E rotates the cage until the rollers jam between the housing and the cam. If the cam rotates faster than the housing, the rollers are freed, and the rotation of the cage, relative to the cam is restricted by the stop-pin F. The drive passes through the housing and the rollers to the cam, which is integral with the gear G. In an alternative arrangement that has been prepared, the gear is the driving member.



Right: The components and layout of the new free-wheel assembly

AUTOMATIC TRANSMISSION

FOR those who wish to know something of the features and operation of the Borg-Warner automatic transmission, without delving into technicalities, the company has issued a lively little guide booklet. The information is contained under 64 headings, arranged alphabetically, and is concise yet comprehensive. Copies of the booklet may be obtained free of charge on application to Borg-Warner Ltd., whose address in Great Britain is Letchworth, Herts.

BRAKE LININGS

WE have been informed by Small and Parkes Ltd., of Hendham Vale Works, Manchester 9, that their DON R.14 material is used exclusively for the linings of the brake shoes on the Austin Gipsy, described in the September 1958 issue of *Automobile Engineer*. This has been confirmed by the Austin Motor Company Ltd., who add that the initial specification called for the material quoted in the article, but it was subsequently changed.

A NEW FORD ENGINE PLANT

Full Use of Automation Techniques for a Production Rate of 4,000 Units per Two-shift Day

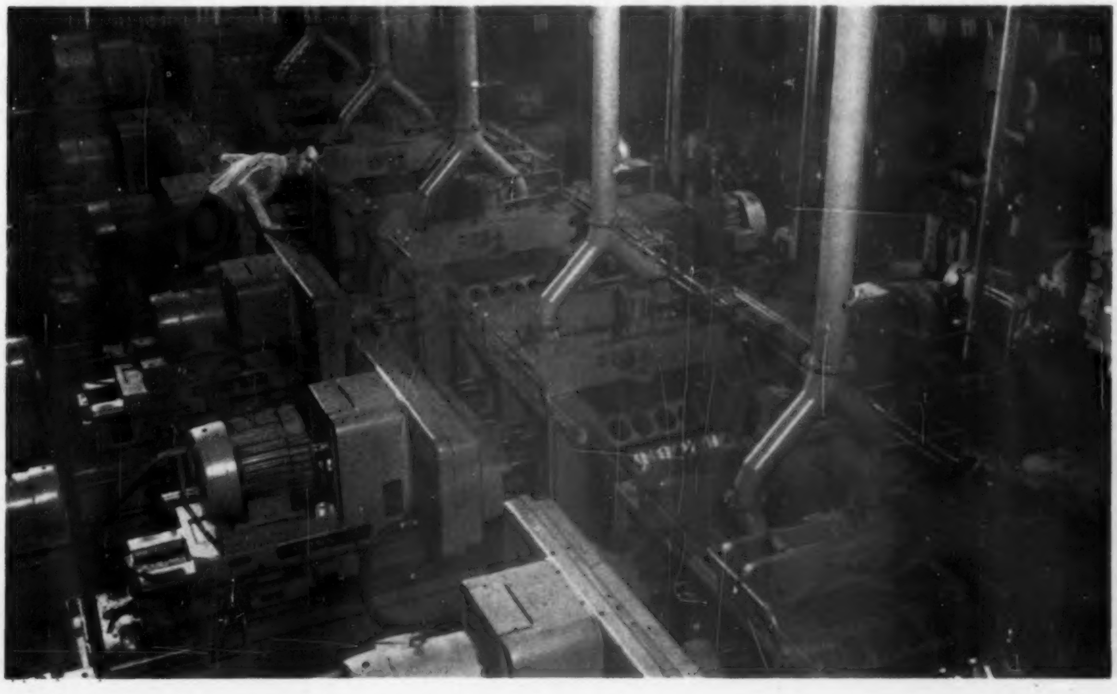
At Lima, Ohio, a new Ford Motor Company plant with a floor area of more than one million square feet is now available for the production of Mercury, Edsel, Lincoln and Continental engines. It was planned and equipped for an output of 4,000 engines daily, on two-shift operation, and this figure can be attained when it is warranted by demand. It is the fourth Ford plant to employ the technique of automation in the manufacture of automobile engines. The Company's Cleveland Engine Plant No. 1 pioneered the use of automation seven years ago and was followed in 1953 by a completely re-organized and equipped engine plant at Dearborn, Michigan. Cleveland Engine Plant No. 2 went into production in 1955.

As at other Ford plants, every phase of production where automation has been introduced was subjected first to thorough study and analysis. To be worthwhile the installation had to produce favourable results in terms of improved quality, higher overall efficiency, better working conditions or greater safety. Ford engineers, as a result of the constant study of automation, have developed a comprehensive in-process quality control programme. This method of checking component parts between machining operations has, it is claimed, been developed to the highest degree in the automotive industry at the new Lima plant. Automatic multiple-station checking gauges have been integrated into production lines where they keep a continual check on machined parts to determine that they meet the high standards established by engineering specifications. In addition to the in-line inspection mechanisms, there are approximately 1,700 different gauging applications in use at the Lima plant to ensure the production of engines of consistent quality.

One of the more outstanding automation developments in the plant is the crankshaft area, where crankshafts weighing 80 lb move with ease and precision down one of three 588 ft long machining lines. The crankshafts are automatically loaded and unloaded at the various stations, and are lifted, rotated and angularly positioned. Multiple-station control gauges are built into many of the machines and the part is thoroughly inspected before it passes from one operation to the next. If the crankshaft does not meet specification at any stage, the gauge automatically rejects it and production is halted. The malfunctioning area is indicated on the control console and tooling corrections must be made before machining operations can be resumed. The crankshafts are also balanced by electronically controlled machines which test their own work, and if not satisfactory, automatically repeat the operation. An out-of-balance crankshaft, it is claimed, cannot be passed out from the electronic unit.

A number of operations can be combined in a single in-line transfer-type machine where formerly a series of

Thirty-two separate drilling and reaming operations are performed on an engine block as it passes through this automatic machine. Extraction ducts over the centre of the line remove all cast iron dust from the area



single machines, linked together by various transfer devices, was used. These new in-line machines tend to improve the quality of the work as a greater number of tools can be concentrated and accurately set up in one machine, reducing the number of transfers and indexing and clamping operations. Chip and swarf removal operations are simplified as chips fall into conveyors beneath the machines and are carried away to a central disposal system.

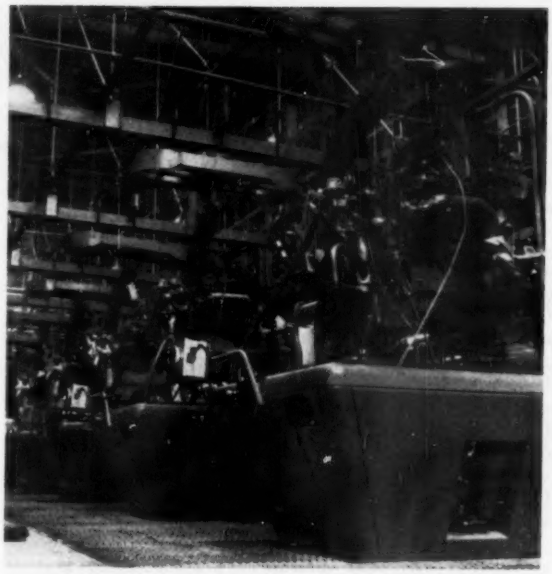
Another automation feature in the new plant is the extensive use of proximity switches to operate transfer equipment. The switches are relatively trouble free as they never come into direct contact with the machined part. Instead, the part passes over the switch and when a close tolerance is reached, the switch activates the equipment. All automation equipment in the plant is hydraulically operated.

The processes of automatic lubrication, scrap removal and inter-operation dimensional checking have been integrated with the lines along with automatic transfer devices. Everything possible has been done to eliminate the manual loading and unloading of parts.

Cold extruded gudgeon pins

The cold extrusion process for the production of gudgeon pins, used for the first time in the automotive industry at the Company's Cleveland Engine Plant No. 2, has been further developed at the new engine plant at Lima, resulting in considerable material saving. Small steel slugs are coated with phosphate and a liquid-soap lubricant and automatically positioned in the press. Carbide punches, mounted on two opposing arms, strike the ends of the slug with 100 tons pressure. The force extrudes the cold metal into the dies, forming the gudgeon pin. The pin is struck again, this time by flat punches to square the ends, and is then automatically ejected from the press. When extruded, the pin measures 3½ in long. The press is capable of producing over 1,900 pins per hour.

Prior to development of this method, and the necessary press equipment, it was customary to drill out the interior of a bar to form a gudgeon pin. This practice was time-consuming and resulted in considerable waste of valuable material. The only drilling operation necessary on an extruded gudgeon pin is to bring the pin within weight tolerances. The pins also pass through heat-treatment furnaces and through automatic grinding machines which finish the outside diameter. An electronic instrument, located in a temperature-controlled room, is used for the precision inspection of all pins before they are sent to assembly areas. The pins are fed into the machine and are sorted into three different classifications according to minute differences in size and weight.



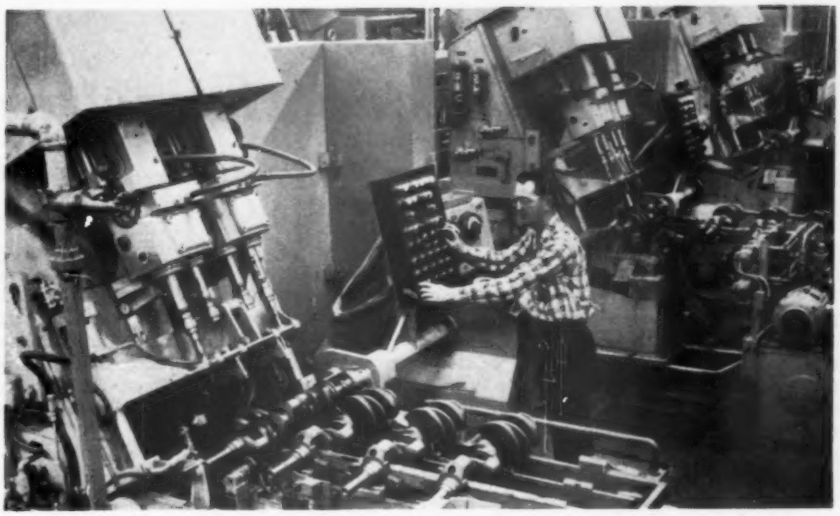
Every engine produced is finally tested on one of the 96 engine test stands before being passed out for dispatch

Other outstanding features of the Lima plant include the use of electronics to correctly balance all engines; a Tele-Autograph system which makes possible precise selection and rapid fitting of engine parts along the assembly line; and a preventive maintenance programme.

As a cylinder block approaches the beginning of the assembly line its dimensions are checked and transmitted by the Tele-Autograph to various locations in the plant where engine components are selected to match the cylinder block dimensions. The components are placed on conveyers for delivery to engine assemblers and reach the assembly area shortly before the matching engine block arrives.

The assembly area includes two lines, each composed of 31 independent conveyor units coupled together like links

This completely automatic transfer machine can handle 600 pistons per hour. It performs seven different drilling, reaming and milling operations



Crankshafts are balanced on these electronically controlled machines. The machine determines how much the crankshaft is out of balance, where the unbalance is located, and then drills counterweights to correct. When perfect balance has been attained the crankshaft is ejected from the machine

in a chain. This system makes it possible to continue production along the line even though one section of the line may be down for repairs. Engine blocks are suspended from the overhead monorail tracks on steel-arm carriers. No less than 96 test stands are used for the final testing of all engines before they are dispatched to assembly plants throughout the country.

Maintenance

The advent of long, fully automatic production lines has led to the need for communication along the lines. This problem is particularly acute when a breakdown occurs near the end of a line and the control console which indicates the performance is located at the beginning. A maintenance man must leave his work to check at the control console, resulting in considerable delays. The telephone has been brought in to solve this problem at the Lima plant.

Telephone equipment is installed on the cylinder block front cover line, for example, which is approximately 70 ft long and has 26 different stations for machining the die-cast aluminium parts. A telephone set is provided at the start of the line where the operator and the control panel are located and telephone sockets were fitted at all stations along the line. When a breakdown occurs the operator directs the maintenance man to the trouble area, where he plugs in his portable set and remains in constant touch with the operator and the control console while making necessary adjustments. When repairs are completed, he advises the operator to start production once again.

The telephone also works as a two-way device. When regular preventive maintenance tool changes are scheduled, a maintenance man can plug in his set anywhere along the line and notify the operator to shut down the equipment. He quickly makes his changes and then informs the operator that the line can be started again. Use of the telephone system has enabled the plant to keep the down time of the lines at a minimum.

A modern fresh-air distribution system brings about a complete change of air in the plant every eight minutes. Air diffusers come to within 10 ft of the floor to allow for a quick and even distribution of fresh air.

Numerous fire prevention devices have been provided throughout the plant. Giant smoke and heat-venting hatches that open automatically, fire curtain boards, automatic sprinklers and a specially constructed roof all play important roles in the prevention and fighting of fires.

Commercial Vehicle Flooring

A Polyester-Glass Fibre Covering Developed to Minimize Wear and Damage

LOADS of an abrasive or heavy nature frequently have to be carried by commercial vehicles and result in rapid deterioration of the flooring. Among the worst offenders in this respect are coal, coke and granite. To lengthen the life of floors on which such loads have to be carried, a jointless floor covering has been evolved by Tom Byatt (Engineers) Ltd., Fenton, Stoke-on-Trent. It is based on Bakelite polyester resin and glass fibre mat, and the method of construction has been approved by the Board of Trade for use in calibrated bodies.

First, the original floor of the vehicle is scored to provide a key, after which a layer of resin, suitably modified by the addition of abrasive grit, is spread on it in a continuous layer. Over this coating is applied a 2 oz, chopped-strand glass mat, which is impregnated with a similar resin mix. The final layer comprises the same resin but with a higher proportion of the grit. After consolidation by rolling, the laminate is allowed to set hard, and then the treated body is ready for use.

The Byatt flooring is claimed to be suitable for any type of loads except those that might burn the resin, for example, hot ashes. Since the material is waterproof, it can be employed in vehicles that carry wet loads, such as sand or gravel. The covering is suitable for use in tipper bodies, because it has an unbroken surface and therefore offers no resistance to unloading. Further advantages are ease of repair, should this become necessary, and low weight: the Byatt floor covering for an average 6 ton, long-wheelbase vehicle is said to weigh about 36 lb, as against 1½ cwt of comparable steel lining used for floors of this type.

Precision Barrelling

The Development of New Methods and Processes by Rolls-Royce Ltd.

UNDER a variety of names such as rumbling, tumbling, slide-abrating, and rotary finishing, what is now generically termed barrelling is undoubtedly one of the earliest finishing processes. As with other mechanical innovations, its origin cannot be established with any certainty. The basic principles of the process were, however, well known to medieval silversmiths and cutlers and were widely applied by them in the polishing and burnishing of their wares. Metal products to be treated were placed in a small wine cask, fitted with a door and an axle and crudely mounted for rotation, which was turned by the hands or the feet of juvenile labour.

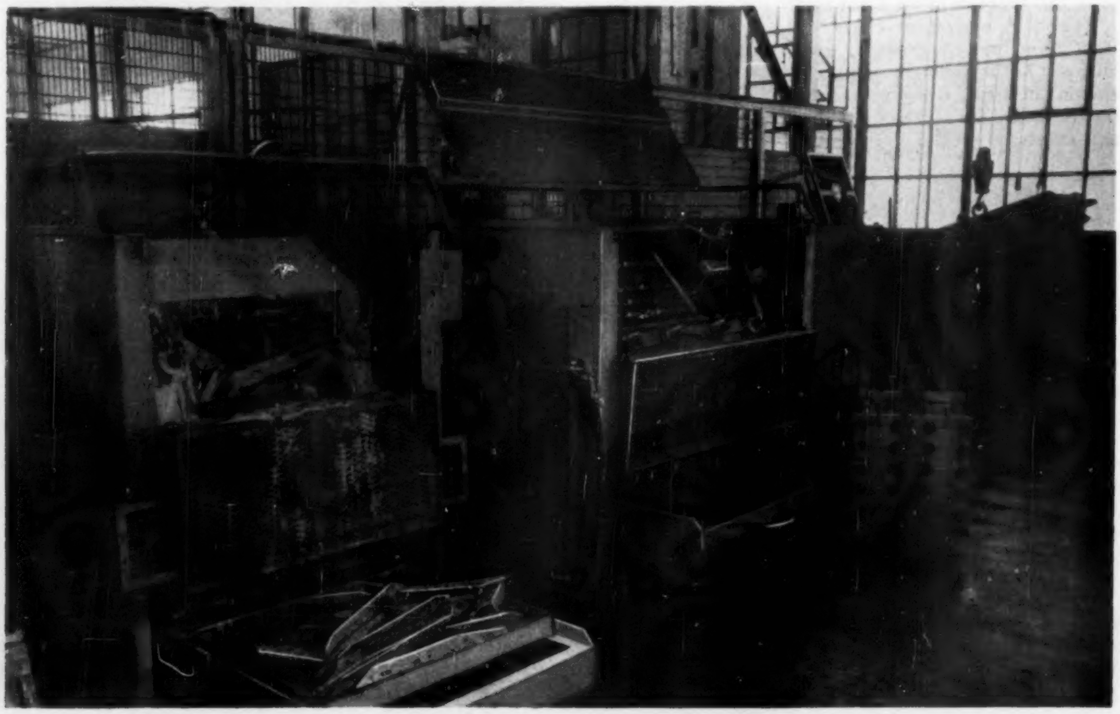
Some items were merely tumbled by themselves, without any added materials, but commonly leather scraps and clippings were included to enhance the burnishing effect. Leather clippings were eventually replaced by other media. During the evolution of the process it was found that various combinations of beach pebbles, sand, and water successfully removed burrs and surface defects from metal parts, thereby increasing the scope of the process beyond simple burnishing. Beach pebbles were succeeded by granite and other stone chips and these were displaced in relatively recent times by synthetic chips prepared from fused aluminous oxide. These chips possessed the advantages of greatly increased abrasive properties and consistency in operation but their rate of depreciation was more rapid than that of granite chips.

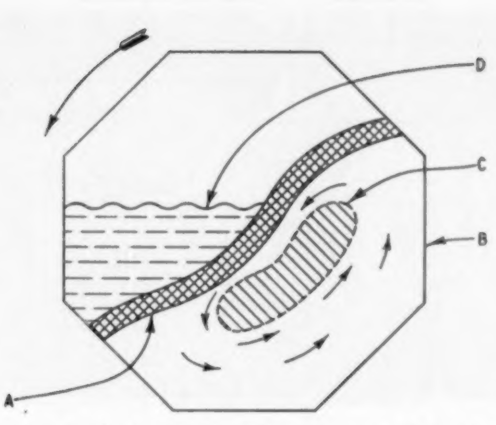
By that time the wine cask had been replaced by a circular or hexagonal barrel, commonly belt-driven from a line-shaft, and this in turn by the modern, rubber-lined hexagonal barrel with individual electric drive. In combination, these chips and barrels enabled the process to be developed from a rather crude operation dependent upon the care and skill of the operator to a reasonably predictable and timed machine operation controlled by unskilled labour.

Preformed Shapes

At the end of each barrelling cycle, the separation of processed workpieces from, possibly, several hundred-weights of chips presented a problem difficult of solution. Various methods, including magnetic separation, were evolved and were successfully applied. The most common was the vibrating or oscillating wire mesh screen. Where workpieces and chips were of similar size, however, the irregular shape of the chips frequently defeated the best of screening efforts. A further problem arising was the lodgment of chips in holes, slots or other apertures in the workpiece: it was difficult to correlate the size of chips to the work and the screening. In attempts to overcome this problem chips of specific size and shape were produced from mouldable or extrudable materials and termed preformed shapes. Until fairly recently it was impossible with the

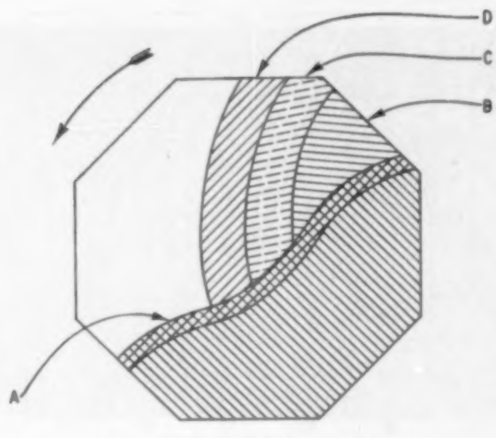
Fig. 1. Battery of Roto-Finish machines at the Crewe works of Rolls-Royce Ltd. At the left is a medium-capacity machine handling the pressings shown in Fig. 4. Camshafts are processed in the two larger machines. A loaded camshaft fixture is standing on the floor adjacent





A effective sliding layer of chips; B returning chips; C dead area; D level of solution

Fig. 2. Barrelling action



A effective sliding layer of chips; B at 20 rev/min; C at 35 rev/min; D at 45 rev/min

Fig. 3. Turnover points

materials and the techniques available to produce, within economic limits, preformed shapes of grinding capacity equal to the fused aluminous oxide chips. Nor was it possible to provide adequate resistance to fracture together with the slow dimensional wear rate essential in preformed shapes. These difficulties, however, have now been overcome. A successful development by Rolls-Royce Ltd., is a preformed triangular shape produced by a special process in which various hydrous silicates of alumina are blended together and combined with selected abrasive grains. While the grinding ability of these shapes is comparable with that of ordinary irregular-shaped fused aluminous oxide chips, their wear rate is approximately only one-third of that of the conventional chips.

Steel balls and cut steel shot are also used for burnishing and for "colourizing" or influencing the brightness of the surface finish of the work. These present similar difficulties as regards separation after barrelling cycles, with the additional disadvantage that except in the case of non-ferrous workpieces, magnetic separation methods cannot be employed. Steel media, however, are also produced in preformed shapes and are frequently used on suitable work.

Sand replaced by compounds

The use of sand as an abrasive in barrelling is today relatively rare. Instead, modern abrasive grains are combined, severally, with brighteners, cleaners, wetting agents, and corrosion inhibitors either in the form of powders or made-up liquids. These mixtures are commonly referred to as "compounds". Coarse-grained abrasives are included where a quick grinding action is required and finish is of a relatively secondary importance. Compounds containing fine-grained abrasives, or, in some instances, water-soluble lubricants capable of reducing the naturally abrasive character of the chips or shapes, are used in finishing or second-operation compounds.

Planning barrelling operations

Many variables need to be taken into consideration when planning barrelling operations. The suppliers of barrelling equipment, grinding media, and compounds are continually investigating and testing materials and methods and their accumulated experience is available to users. When a barrelling operation is under consideration for a specific component it is necessary to ascertain:

1. Type, strength, hardness and toughness of the component material
2. Work to be done on the component
3. Surface finish required
4. The "colour" or brightness to be produced
5. Limiting factors affecting edge radii, overall dimensions, bores and projections
6. The subsequent treatment to be applied—machining, grinding, plating or painting

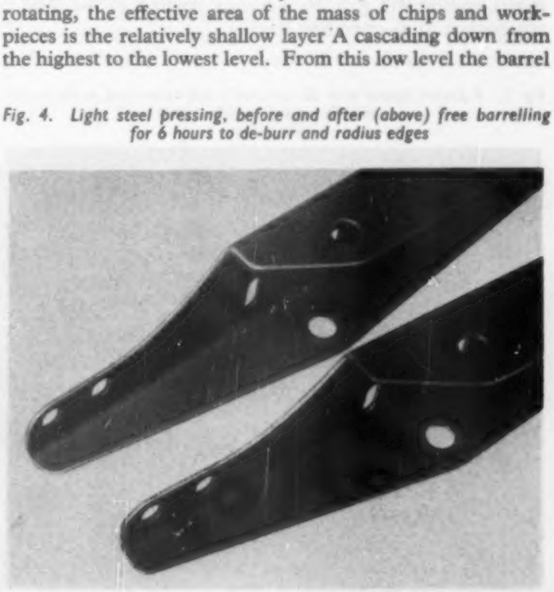
When these facts are known it is possible to decide whether a single operation or sequential operations are required to obtain the required result. Consideration can then be given to the conditions in the barrel, for one or more operations, to obtain the desired result.

For each operation it is necessary to determine the following variables:

1. Type and size of the media
2. Work load or number of workpieces
3. Mass level in the barrel
4. Water level in the barrel
5. Type and quantity of compound
6. Speed of the barrel
7. Process time

Reference to the illustration diagrammatically depicting barrelling action, Fig. 2, will make clear the importance of several of these items. At any instant, while the barrel is rotating, the effective area of the mass of chips and workpieces is the relatively shallow layer A cascading down from the highest to the lowest level. From this low level the barrel

Fig. 4. Light steel pressing, before and after (above) free barrelling for 6 hours to de-burr and radius edges



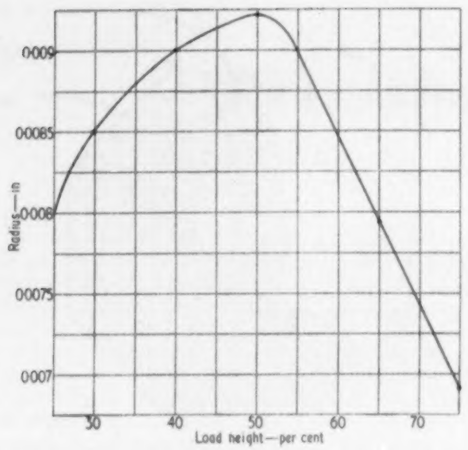


Fig. 5. Effect of barrelling at various load heights; 4 in × 4 in × ½ in steel plates for 4 hr

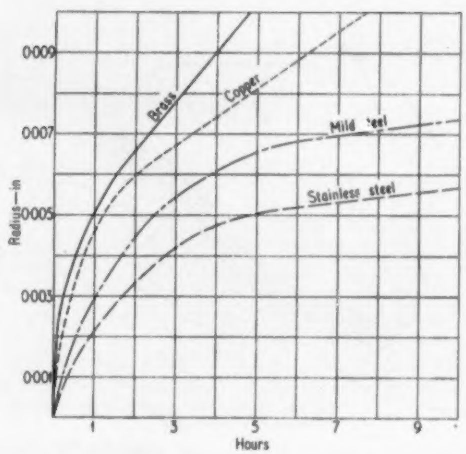


Fig. 6. Effect of barrelling time on radius produced; on brass, copper, mild steel and stainless steel

elevates chips and work to the high level where they continuously break loose from the mass and slide downwards. Movement is the more pronounced in the peripheral layers but in the centre there is a more slowly moving, but continually changing, mass where little effective work is performed. This is termed the "dead" area.

In practice, the mass level may be varied from centre level, 50 per cent load height, to almost seven-eighths full, say 85 per cent load height. The higher the mass level the shorter and slower will be the effective slide. This gives a less drastic action, resulting in slower processing but yielding improved surface finish. Conversely, the lower the mass level, to centre height, the higher the speed in the slide, giving faster processing but poorer surface finish. The effect of variation of load height is shown in the curve, Fig. 5. This indicates the edge radii produced on steel plates processed at different heights, all other conditions, of course, being constant. It shows conclusively that, for maximum metal removal, the most effective height is 50 per cent. Before checking mass level, the barrel should be rotated and the contents allowed to settle. To check on initial loading will give an incorrect reading as the mass is not packed to its operating level.

The water or solution level may be from 1 in above the bottom of the barrel to completely full. Its influence is similar to, but less pronounced than, that of the mass level.

In connection with this sliding action, the rotational speed of the barrel is an important factor. Obviously, the higher the speed the faster the slide and consequently an enhanced cutting action but a poorer finish. Even in instances where surface finish is of secondary or minor importance, however, barrel speeds cannot be raised unduly in order to shorten process time without introducing a new and usually undesirable influence. This is shown in the diagram, Fig. 3, of the so-called turnover points. As the speed of the barrel is increased, so the mass of chips and work is carried higher until eventually sliding action on the surface ceases and the discrete mass falls after breaking away at the top of the barrel. Little work is accomplished since abrading action is slight and, instead, the surface of the component being treated may be damaged by impingement of both chips and other components.

Processing time will depend on the condition of the work as received, the size of the burrs or flashes to be removed, the chips and the compounds used, and on the surface finish required. It may range from 1 hour for a light deburring

Fig. 7. A fixture loaded with 20 camshafts and assembled in the barrel, ready for charging with media

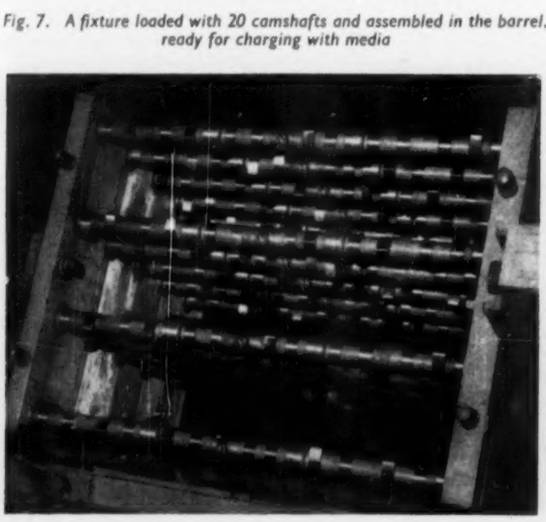
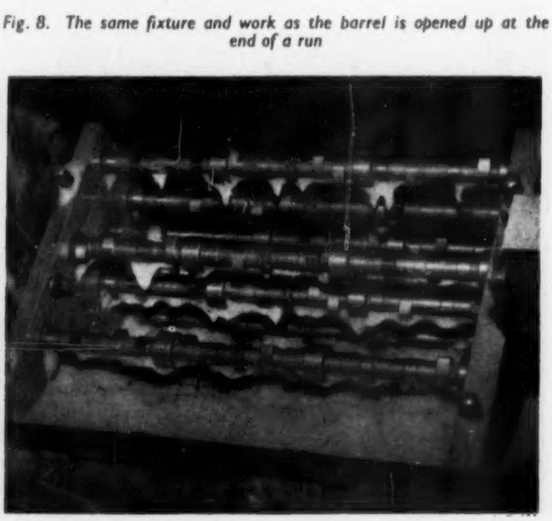


Fig. 8. The same fixture and work as the barrel is opened up at the end of a run



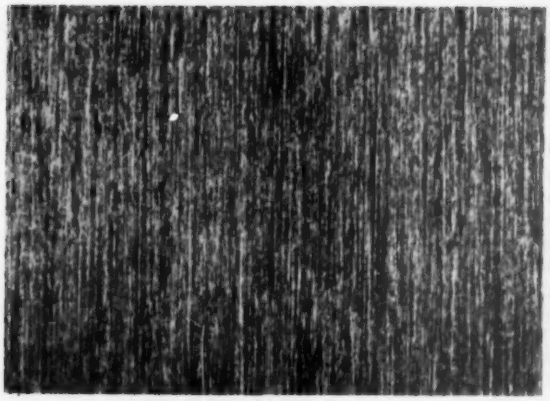


Fig. 9. Micro-photograph of the normal fine-ground surface finish of a cam profile, × 30

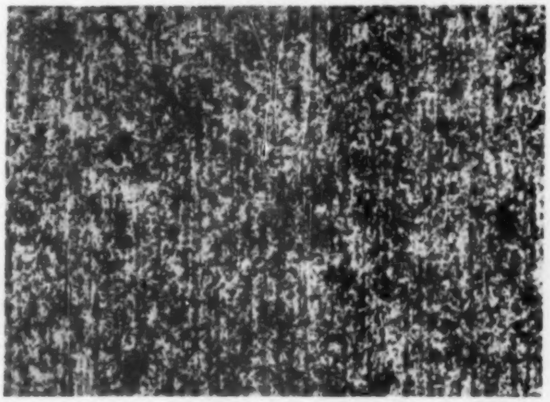


Fig. 10. The same profile after barrelling treatment to give an oil-retaining surface finish, × 30

operation to as much as 20 hours for a heavy grinding or a superfine polishing process. When initially processing a new component it is usual to check on the suitability of the selected routine after a short run and then to examine the work at regular intervals until a satisfactory result is obtained. Typical times for edge radiusing four different metals—brass, copper, mild steel and stainless steel—are given in the curves of Fig. 6.

All process data relevant to an approved treatment for a particular component is recorded for subsequent use on further batches of the same component. In this manner, it is made possible for the process to be repeatable. When work is put in hand, a process card prepared from the records is issued to the barrelling operator. This lists the work load or quantity, process time, type of media, mass level, type of compound, quantity of compound, water level, and barrel speed, and includes any special instructions.

Setting up the barrelling machine for a run thus becomes a straightforward procedure, requiring care in the use of the specified materials and reliability in connection with the measuring or weighing of the compounds. By a new processing method introduced by Rolls-Royce, the operator will be relieved of even this modest responsibility. It is claimed to be virtually foolproof, while at the same time it overcomes certain barrelling problems which hitherto were regarded as insoluble.

From loose compounds to tablets

In this recent Rolls-Royce development selected short-life, coarse grains are combined with long-life, fine grains, lubricants, inhibitors, and brighteners in the form of tablets. These tablets have a controlled rate of disintegration, and can provide the rapid grinding of components and a good finish in a single operation. Thus, by eliminating the necessity of second or finishing runs, overall processing time is much reduced and throughput is increased. The tablets are manufactured in seven grades and are distinctively coloured for ready identification. While each grade has a working range much wider than an ordinary abrasive compound, seven grades will not cover all possible applications on a wide variety of materials. Accordingly, the formulae have been compounded and the tablet disintegration times so arranged that a combination of grades of tablets can be used simultaneously to attain a desired result. Data is available of suitable combinations of tablets designed to meet specific barrelling problems on all metals and alloys in general engineering use.

Barrelling equipment installed at the Motor Car Division of Rolls-Royce Ltd. at Crewe, was supplied by several manufacturers and is of widely varied capacity. A number of different barrelling methods are used, selectively employed as best suited to produce the desired results on particular components. Preceding comments have all related to "free" barrelling, in which the workpieces are indiscriminately mixed with, and move freely in, the mass of chips in an unobstructed barrel. This method is the most commonly used for small and medium-sized parts of compact shape. A variant of free barrelling is "submerged" barrelling. The work is loaded with the chips into a perforated barrel which is rotated while suspended in a tank of solution. Obviously, this method lends itself to multi-operation processing. The loaded barrel can be lifted out of one solution, drained, washed, lowered into another solution, or into dip tanks for corrosion inhibition, brightening, or lubricating. It provides a means of arranging the barrelling of large quantities of parts on a production basis by means of fully mechanized, accurately timed, automatic transfer equipment.

Fixture barrelling

Relatively large, slender, or thin-section components may be susceptible to damage due to impact if tumbled freely in a barrel. To obviate this hazard, resort is had to "fixture" barrelling. The parts are attached in spaced relationship to suitable fixtures securely mounted in the barrel. A variant of this method is "compartmented" barrelling. As implied

Fig. 11. Fully loaded barrelling fixture for 20 six-cylinder engine camshafts

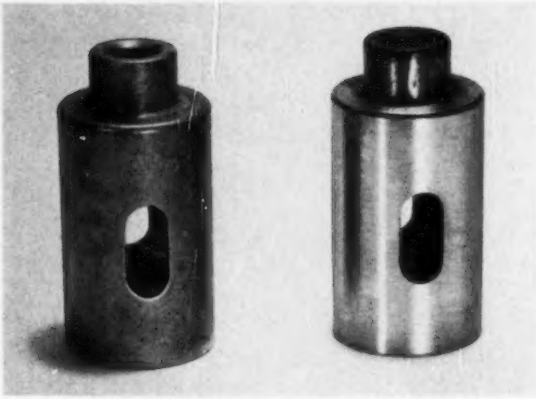


Fig. 12. Engine tappet de-burred and edge-radiused by barrelling (left) and after finish grinding to size (right)

by the name, the barrel is provided with a number of compartments each accommodating a single component or a few components on a simple fixture.

Barrelling camshafts

One of the most interesting operations at Rolls-Royce, Crewe, is the barrelling of engine camshafts to remove sharp edges left by machining and grinding operations, and to impart to previously ground profiles a special oil-retaining surface finish. This is a fixture operation, performed in one of the two large-capacity models in the battery of Roto-Finish machines shown in Fig. 1. Both machines are provided with special large doors to permit a loaded fixture, Fig. 11, to be inserted or, in subsequent runs, for the parts to be assembled on the fixture already in position in the barrel.

The fixture framing, of channel section aluminium alloy, is bolted through spacer blocks to the end cheeks of the barrel. Threaded over long bolts secured to the base of the framing are clamping blocks, also of aluminium alloy, formed at their contiguous faces to accommodate the end bearings of four camshafts. Five banks of shafts are assembled, making a charge total of twenty. The two intermediate journals and the oil pump-timer drive gear are left unmasked, as the treatment enhances rather than impairs their surface finish. On one side, however, the clamping blocks are recessed to receive the timing gear flange and thus to shield the tapped holes in the flange. Figs. 7 and 8 respectively show the charge of camshafts in the barrel before and after processing.

A mixture of large and small chips is employed in conjunction with an abrasive compound and, as is usual in fixture barrelling, barrel rotation is alternated in direction to ensure complete coverage of the work. Rotational speed is 15 rev/min and the total running time is 2 hr, divided into a run of 1 hr in each direction. Before employing barrelling for this work, the labour time allowed per shaft for individual hand frazing was 15 min; on the machine this figure has been reduced to 4½ min. Of course, hand work did not touch or have any effect on surface finish. The utmost importance is attached to this aspect. Due to the wiping action under load of the cam nose on the tappet, this point, potentially, could be a source of trouble arising by lack of lubricant. Careful development of the barrel process has enabled a discontinuous surface finish having a matt appearance to be consistently produced. Fig. 9 is a micro-photograph of the fine-ground surface of a cam before barrelling, and Fig. 10 shows the discontinuous, oil-retaining surface produced by this barrelling technique.

A tangible advantage has been registered for the new surface treatment. It had been observed earlier that in a new engine initial running produced 0·0005 in loss in lift, literally a "bedding down" of the surface. Since the new discontinuous finish was introduced no measurable wear resulting from initial running can be detected.

Another somewhat unusual process developed by Rolls-Royce relates to the lantern type tappet shown in Fig. 12. It is essential that the apertures in the body are free from sharp edges. Hand frazing was a time-consuming operation and not unlikely to lead to inadvertent damage to the finish-ground body. These tappets are now barrelled for 4 hours in a medium-sized machine before final grinding. They are processed in batches of 500, with grinding chips and an abrasive compound, at a load height of 75 per cent. This produces a relatively large radius on the edges of the apertures, as shown on the left of the illustration, and when the tappet is finally ground to finish diameter only part of this radius is removed, leaving a smooth unbroken edge.

One small component well illustrates the value of barrelling for mass production operations. Reference is made to the familiar split cone securing the valve spring retainer to the valve stem. In dividing these cones into halves substantial burrs are left. To remove these burrs by hand was a most tedious and time-consuming operation. At Rolls-Royce these components are made in batches of 30,000 to 32,000 halves. The labour time expended on the hand frazing of a batch was more than 420 hr. In a barrelling operation a batch is processed for 14 hr, arranged on an over-night run when the plant is shut down. The labour time involved is 1 hr for loading in the evening and 1 hr for unloading the following morning. The result is of good quality and of a consistency not attainable by hand methods.

Rolls-Royce Ltd. pursue an active policy of development of barrelling technique under the guidance of Mr. Smith-Gorman, at Hillingdon, Glasgow, and Mr. Matthews at Crewe.

INDUSTRIAL SAFETY BOOKLETS

A NEW series of booklets on Industrial Safety, Health, and Welfare subjects is being published by H. M. Stationery Office on behalf of the Ministry of Labour and National Service. Two of these, Nos. 1 and 3, are now available.

The first is "Lifting and Carrying", which explains in clear language the right and wrong ways of handling loads and draws attention to common faults. It is well illustrated with photographs and outlines the principles of lifting, which depend on the skilful use of the correct muscles rather than on physical force. Some 50,000 accidents per annum are caused in the handling and carrying of goods, and this booklet should help to reduce the figure. The cost is 1s. 0d., and a brightly coloured wall sheet to supplement the impact of the booklet can be obtained from the same source also for 1s. 0d.

Booklet No. 3, entitled "Safety Devices for Hand and Foot Operated Presses", states that it is too often wrongly assumed that since they are not powered, these presses are not dangerous. In one year, however, there were 691 accidents on hand presses and 231 on treadle presses. Of these, 83 per cent happened to women and girl operators and approximately 20 per cent were sufficiently serious to be likely to result in permanent injury. The booklet, costing 2s. 6d. outlines the risks and describes the various guards and safety devices that can be employed to prevent accidents.

CURRENT PATENTS

A REVIEW OF RECENT AUTOMOBILE SPECIFICATIONS

Liquid-cooled brake

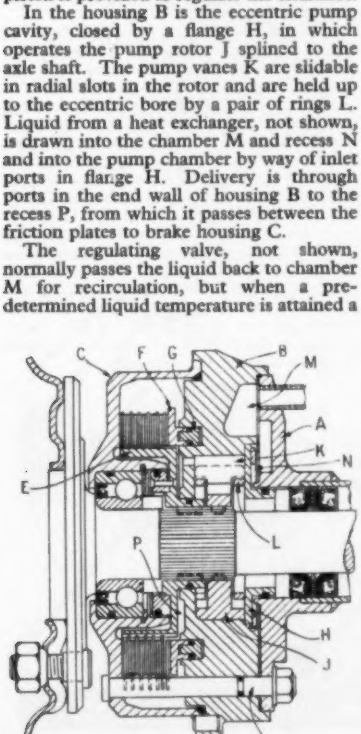
The friction surfaces of this multi-plate brake are directly cooled by liquid circulated by an incorporated, vane-type pump. As the liquid temperature rises, a thermostatically controlled valve brings a remotely sited heat exchanger into the liquid circuit.

A brake flange A is socketed and welded on the end of a rear axle casing and to this is registered housing B embodying the pump unit and, in turn, housing C carrying the sealed ball-bearing supporting the flanged end of the axle shaft. The assembly is secured by eight sealed bolts D. In the annular chamber in housing C is located the multi-plate brake, the stationary plates being engaged on bolts D and the intervening rotating plates being splined to the periphery of a carrier member E splined to the axle shaft.

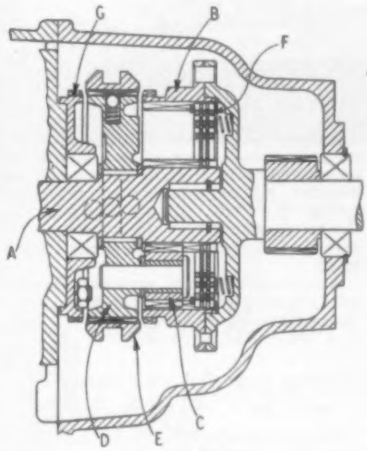
Presser plate F has a grooved flange which projects into a groove in the housing B, where it is engaged by the annular actuation piston G. The clearance space between the piston G and the bottom of the groove forms the wheel cylinder receiving pressure fluid from the master cylinder of the hydraulic braking system. A series of helical springs, not shown, mounted between the wall of housing C and peripheral lugs on the presser plate normally retain the brake in the released condition and an adjuster screw bearing against the piston is provided to regulate the clearance.

In the housing B is the eccentric pump cavity, closed by a flange H, in which operates the pump rotor J splined to the axle shaft. The pump vanes K are slidable in radial slots in the rotor and are held up to the eccentric bore by a pair of rings L. Liquid from a heat exchanger, not shown, is drawn into the chamber M and recess N and into the pump chamber by way of inlet ports in flange H. Delivery is through ports in the end wall of housing B to the recess P, from which it passes between the friction plates to brake housing C.

The regulating valve, not shown, normally passes the liquid back to chamber M for recirculation, but when a predetermined liquid temperature is attained a thermostatic device lifts the valve and allows the liquid to be returned to the heat exchanger. *Patent No.* 792287. *General Motors Corporation* (*U.S.A.*).



No. 792287



No. 793285

Reversing mechanism for torque-converter transmission

On vehicles equipped with a hydraulic torque converter transmission the engine can be idled, while the vehicle is stationary, with the turbine element in driving connection with the road wheels. The turbine element is subjected to a torque which is incapable of driving the vehicle but, nevertheless, is sufficient to create difficulty when effecting engagements. This is especially the case when reversing and accordingly a reversing mechanism is provided, incorporating a friction device that holds the turbine stationary whilst the engine is idling.

The reversing mechanism, embodied in the transmission, consists of an epicyclic train in which the sun wheel is formed on shaft A, from the turbine, the annulus B is fast with the driven shaft, and the planetary pinions C are mounted on a carrier D, freely rotatable on shaft A. Encircling the carrier in toothed engagement is a grooved slidable sleeve E. Translational movement of the sleeve by means of a fork, in one direction engages with teeth on annulus B, locking the planet pinions and giving forward drive through the continuously spring-loaded, multi-plate friction device F. For reverse drive the sleeve is moved in the opposite direction and engaged on member G, secured to the housing, to hold carrier D stationary and drive annulus B, through the planetary train, in the reverse direction.

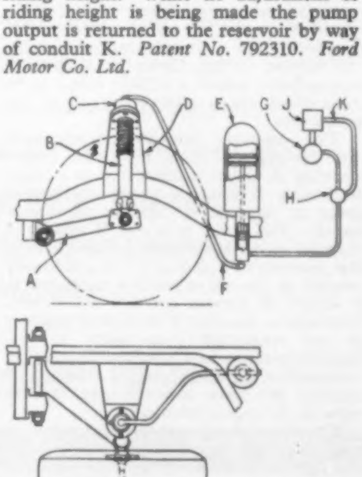
When the vehicle is stopped the friction device holds the shaft A and carrier D stationary, and the sleeve can be easily engaged on member G. In reverse gear, shaft A and annulus B rotate in opposite directions, overcoming the friction of the device F. It is claimed that to slip the plates of device F in reverse is of no material disadvantage since it is only for short periods at low speeds. *Patent No.* 793285. *Regie Nationale des Usines Renault* (*France*).

Hydro-pneumatic front wheel suspension

Means of adjusting the riding height of the vehicle to accommodate increased loading or to give a greater road clearance are embodied in this independent front wheel suspension. The main suspension member is a trailing arm A pivotally mounted on a transverse shaft journalled in brackets on a frame cross member. At its rear end the arm is ball-jointed to the wheelhead, in which is socketed the combined hydraulic shock absorber and telescoping control member B. An annular resilient mounting C supports the upper end of member B from a bracket D attached to the vehicle frame side rail, permitting the necessary angular movement as the wheel rises and falls. Details of the shock absorber and its valves are described in an earlier patent, No. 779597.

The upper, reservoir end of the shock absorber is in continuously open communication with the lower end of a two-diameter cylinder E, attached remotely on the vehicle frame, by way of conduit F. Pistons rigidly coupled by a piston rod are reciprocable in cylinder E, the larger, uppermost end of which contains air at a pressure built up to a predetermined initial value. During wheel bounce fluid is transferred from shock absorber B to the base of cylinder E against increasing air pressure in the upper end of the cylinder. Conversely, during wheel rebound, air pressure returns the fluid to maintain the shock absorber in the fully charged condition.

Cylinder E is also in communication with a hydraulic pump G. A valve H, which may be manually controlled, enables fluid to be supplied to cylinder E to increase the vehicle riding height or transferred from cylinder E to reservoir J to lower the riding height. While no adjustment of riding height is being made the pump output is returned to the reservoir by way of conduit K. *Patent No.* 792310. *Ford Motor Co. Ltd.*



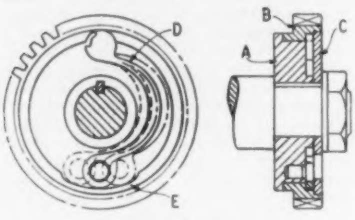
No. 792310

Resilient gears

By reason of their constructional features, resilient driving devices are commonly limited in their application to relatively large gears. A primary object of this invention is to provide a resilient drive gear that does not materially exceed the dimensions of a rigid gear, and can thus be used in confined spaces. It is particularly, but not exclusively, intended for engine timing trains and for governor driving gears.

Referring to the illustration, the gear comprises a hub A, on the reduced periphery of which is mounted for free rotation a ring gear member B. This is retained by a cover plate C which, like hub A, is keyed to the shaft and secured by a nut and washer. In the annular chamber bounded by the recessed face of hub A and cover plate C is housed a solid, arcuate spring D. One end of the spring is pivotally mounted on a pin in the hub while the other, circular end, is engaged in a semi-circular recess in gear member B.

Limited relative rotation between members A and B is permitted in response



No. 792330

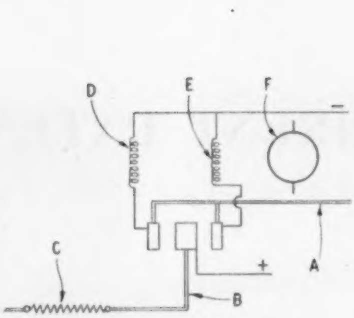
to torque loads and cyclic irregularities, the spring being effective in either direction of movement. An arcuate relief E formed in member B determines the working range by providing positive abutments for the end of the spring. *Patent No.* 792330. *Caterpillar Tractor Co.* (*U.S.A.*).

Maintaining constant vehicle speed

When it is desired to hold vehicle speed to a predetermined constant value, frequent attention must be given to the speed indicator and frequent control adjustments made to the accelerator pedal to compensate for varying resistance encountered—gradient, surface, or wind—as the vehicle proceeds. The automatic control forming the subject of this invention enables a substantially constant vehicle speed to be maintained for a given position of the accelerator.

Two movable elements are envisaged for the control. The first is a forked member A which is connected to a speed indicating device, not shown, for movement in response to changes in vehicle speed. The other is a rod B elastically coupled to the accelerator control through the medium of a spring C. A conductive contact on the end of rod B is connected to a source of electric current, not shown. Contacts on the arms of the forked member A are connected respectively to the oppositely acting field windings D and E of a reversible electric motor F. This motor actuates the usual butterfly-type throttle valve in the engine manifold.

When vehicle speed varies, contact is established between one or other of the contacts on member A and the centrally disposed contact on member B. Motor F is energized and the throttle valve is adjusted to accelerate or retard the engine speed and



No. 793470

consequently to restore the vehicle speed to the determined value. The contacts on member A may displace the contact on member B, compressing or extending spring C. This spring functions in similar manner when the accelerator is moved to increase or decrease the vehicle speed.

An alternative control, actuated hydraulically instead of electrically, is described in the specification. *Patent No.* 793470. *S.A. Andre Citroën* (*France*).

Tipping gear for articulated vehicles

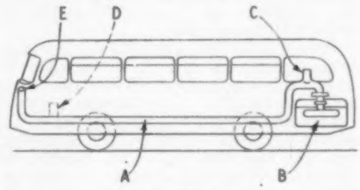
On the tractor-trailer combination illustrated, the forward end of the trailer unit is either permanently or detachably supported on the rear of the tractor unit and articulates about a "fifth wheel". The invention, however, is applicable also to complete, self-supporting trailers and to tracked vehicles. A hydraulic cylinder A for the telescoping tipping gear is mounted on the upper fifth wheel over the axis of articulation and the end of the ram is pivotally anchored to a frame-attached support member B forward of the front bulkhead of the body. Connection of the cylinder to the upper fifth wheel is by means of a substantial eye on the cylinder engaged on a pivot shaft C supported on each side in main and supplementary bearings D and E.

Also mounted on shaft C, between bearings D and E, is a braced, rectangular draft member F. The rear end of this member is pivotally mounted on a transverse shaft G carried between the side members H of the trailer frame. When the forward end of the trailer body is raised to tip the load, relative movement between tractor and trailer units is entailed, bringing their wheels closer together. It is immaterial whether the tractor or the trailer moves. As is the more desirable for the dumping of the load, the brakes of one unit are locked and those of the other unit are released before tipping.

If required, one or more self-supporting trailers equipped with similar tipping gears may be hitched behind the articulated vehicle to form a train. The dumping procedure is then to tip the rearmost unit, draw off and detach, back in again and repeat the operation with succeeding units. *Patent No.* 792094. *The Anthony Company.* (*U.S.A.*).

Damping oscillations in induction pipes

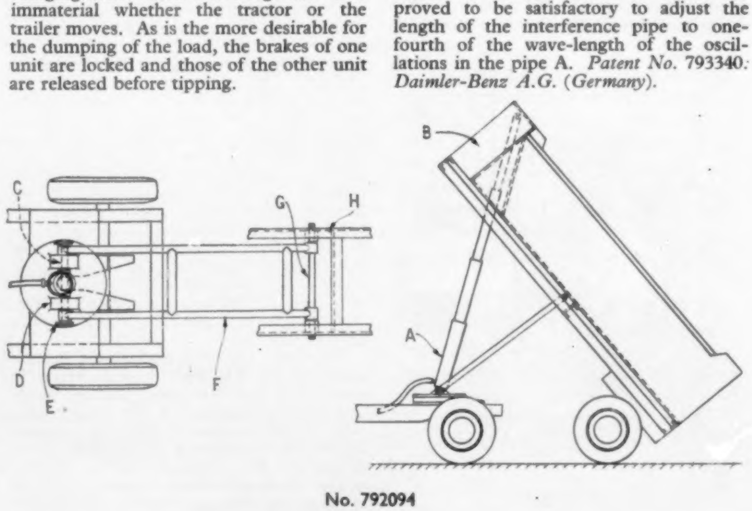
In a vehicle equipped wirh an engine-exhaust brake, disturbing air oscillations may occur in the engine induction pipe when the engine brake is applied. This effect is particularly apparent when the air intake is remote from the engine and the induction pipe is of substantial length, as in the case of rear-engined and underfloor-engined vehicles. Oscillations can be damped, and noise reduced or even eliminated, by the provision of one or more "interference" pipes to the induction pipe. Such interference pipes are branch pipes of suitable length in open communication with the induction pipe and closed at their



No. 793340

outermost ends. Against these closed ends the oscillating air column is reflected to set up damping interference effects in the induction pipe.

A single interference pipe will damp or cancel vibrations of certain frequency but preferably two pipes are fitted. In the example illustrated a single-deck bus or coach has an induction pipe A running from the front of the vehicle to the rear-mounted engine B. An interference pipe C is fitted near the engine, and, if necessary, a second interference pipe D may be provided in the vicinity of the intake orifice E. The required length of each pipe C and D may be calculated or, more simply, be determined experimentally by means of an adjustable closure head. It has been proved to be satisfactory to adjust the length of the interference pipe to one-fourth of the wave-length of the oscillations in the pipe A. *Patent No.* 793340. *Daimler-Benz A.G.* (*Germany*).



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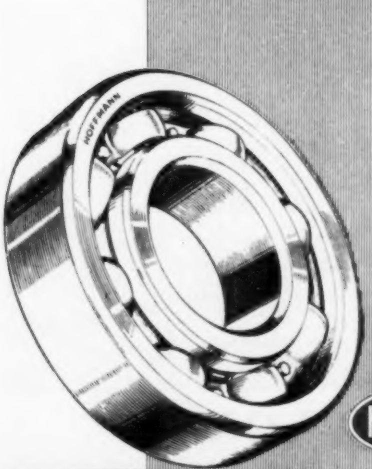

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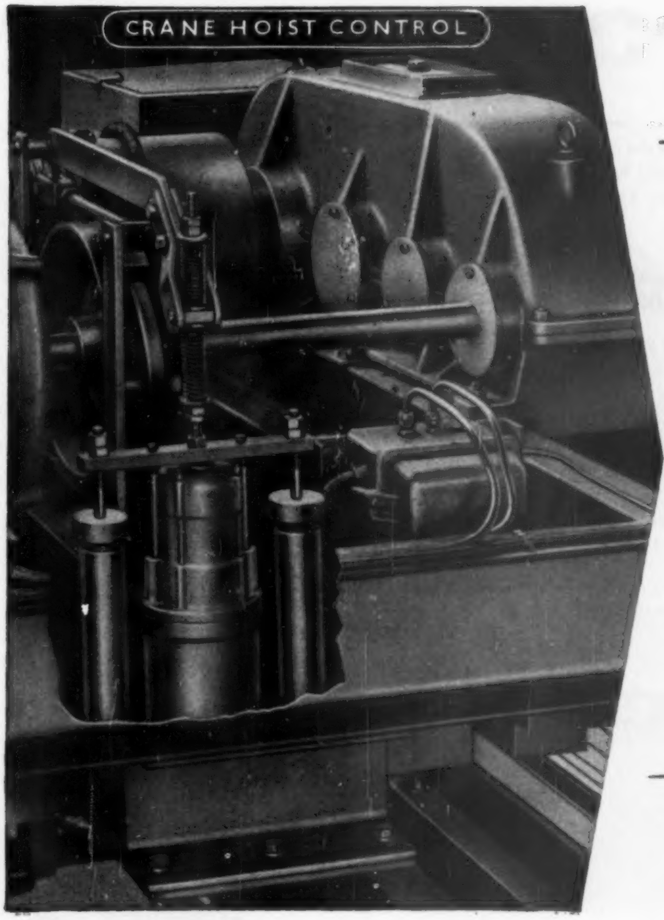

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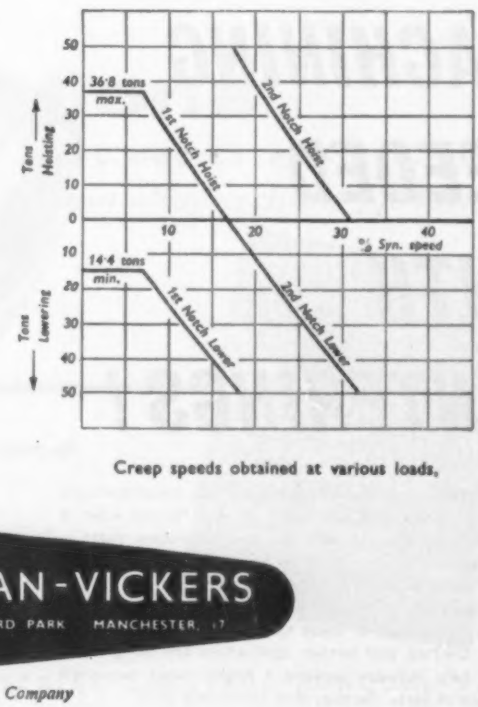

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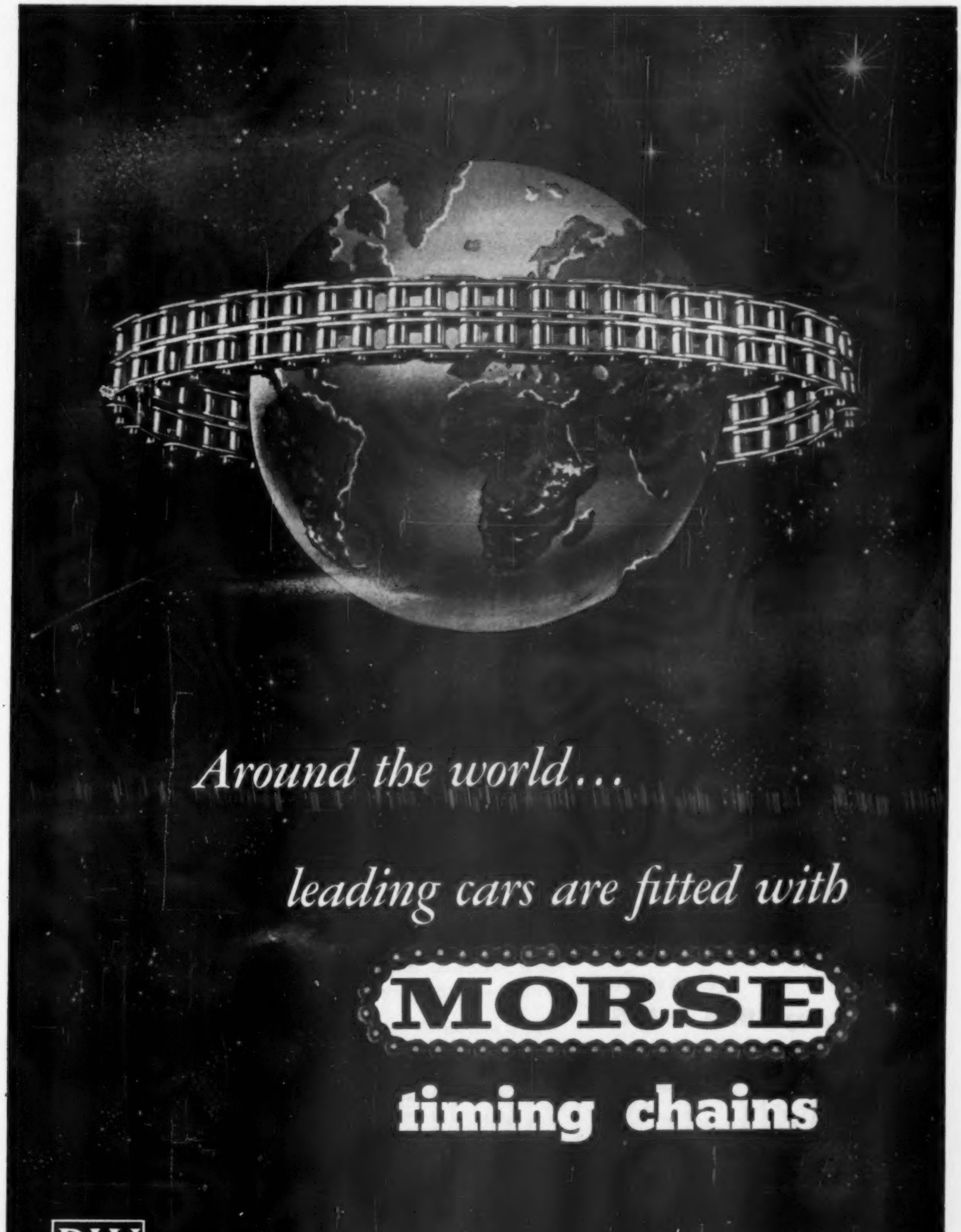

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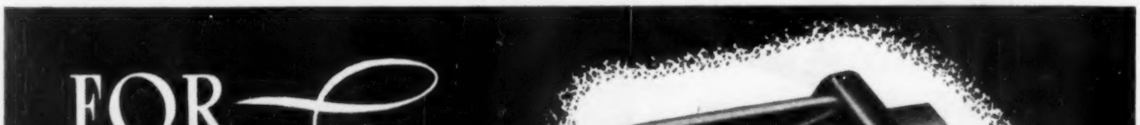

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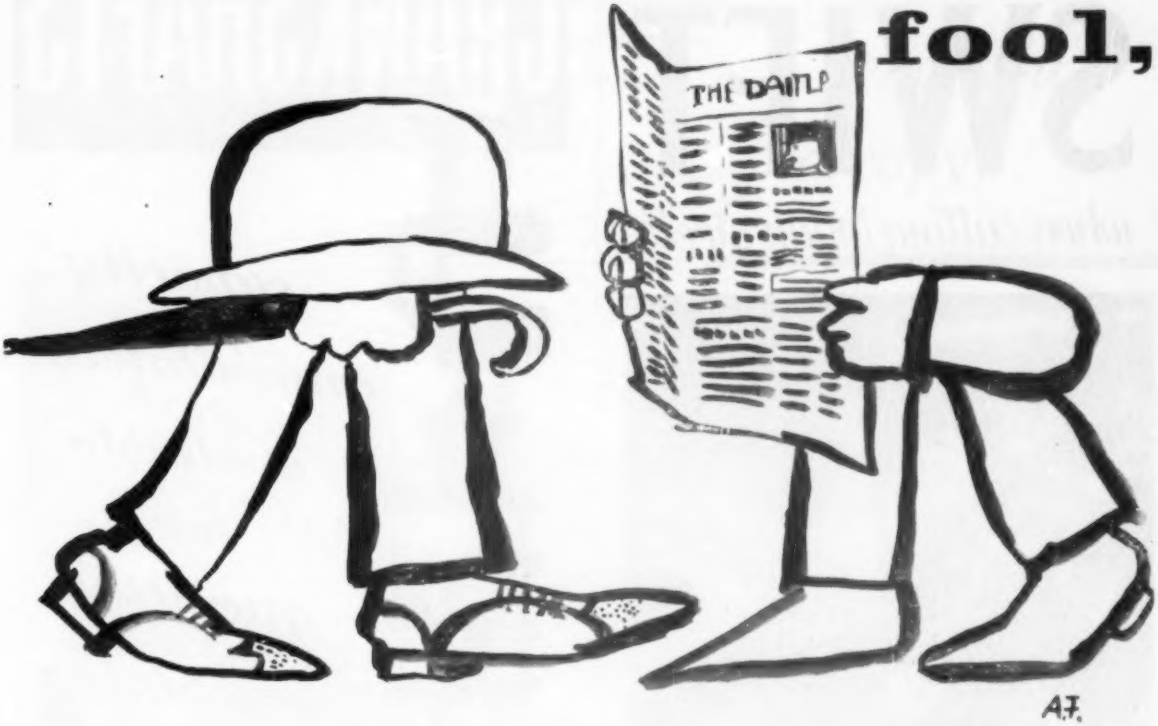

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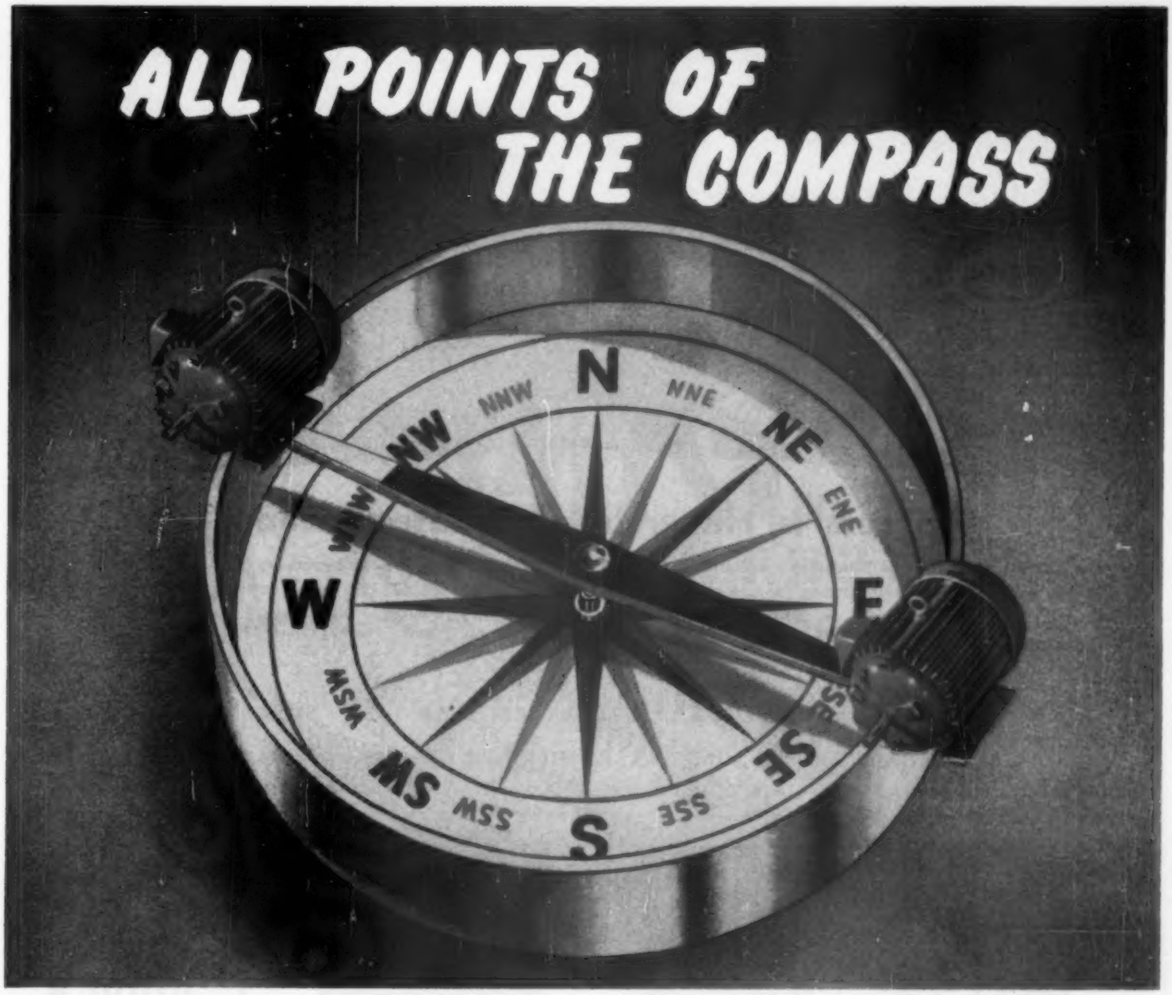
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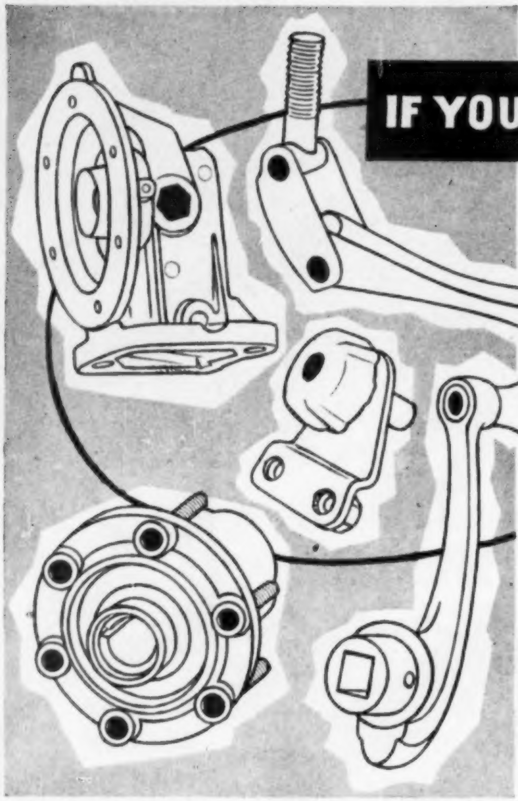

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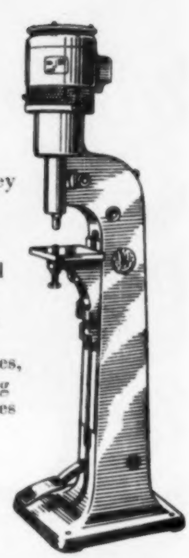

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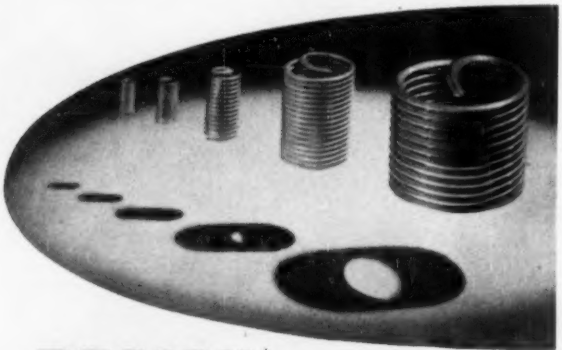

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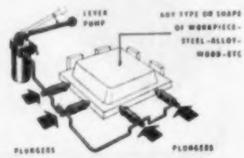

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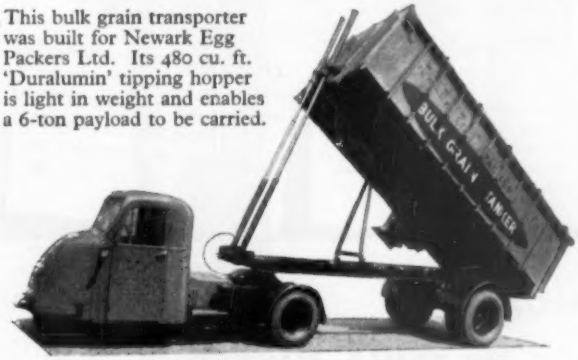

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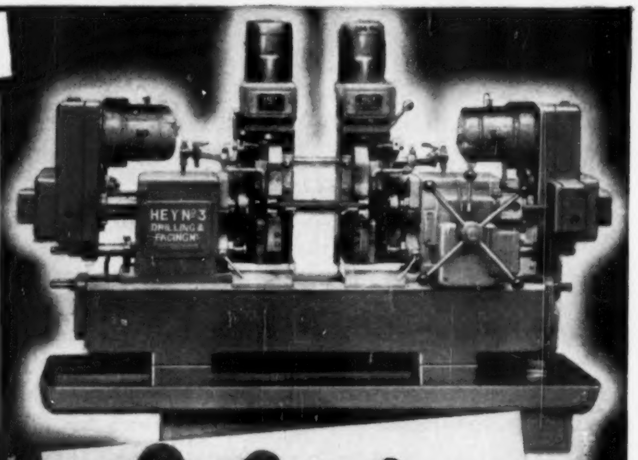

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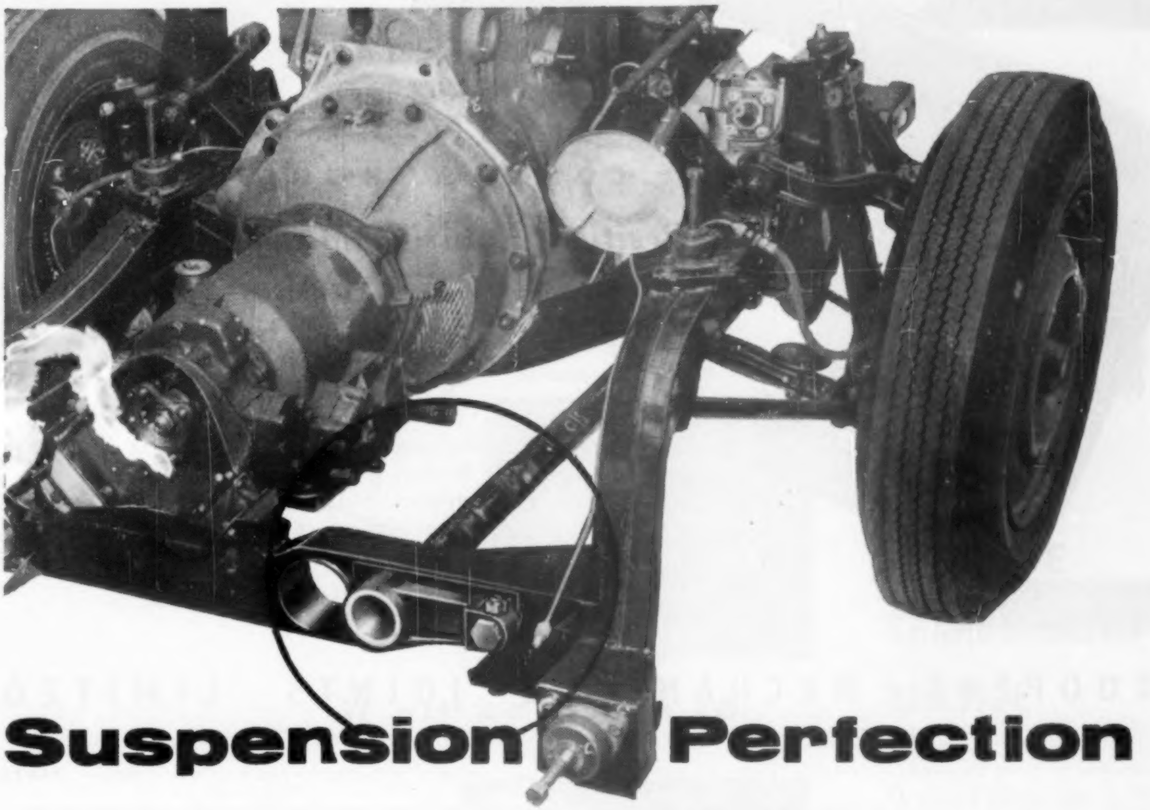

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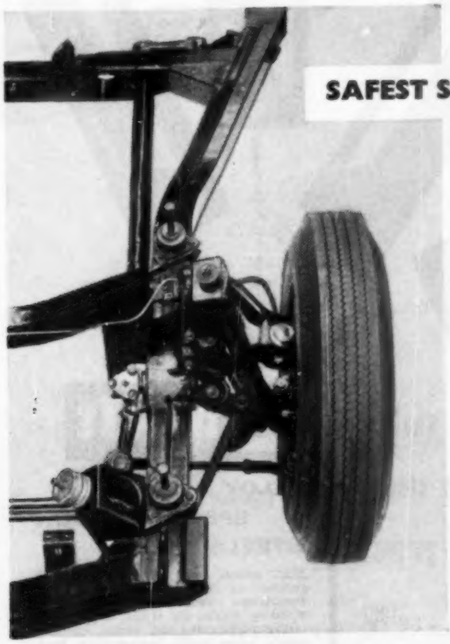

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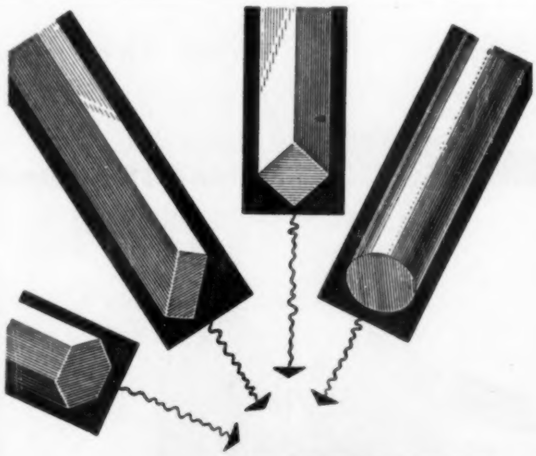

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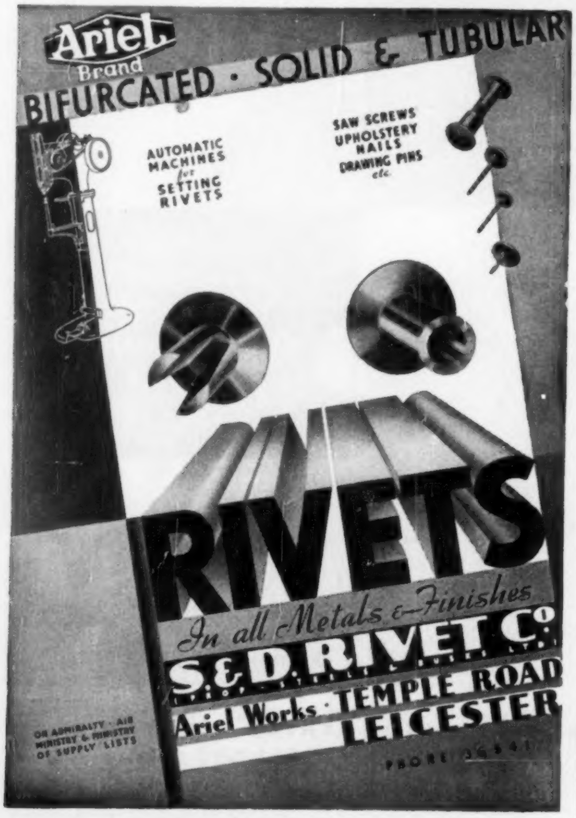

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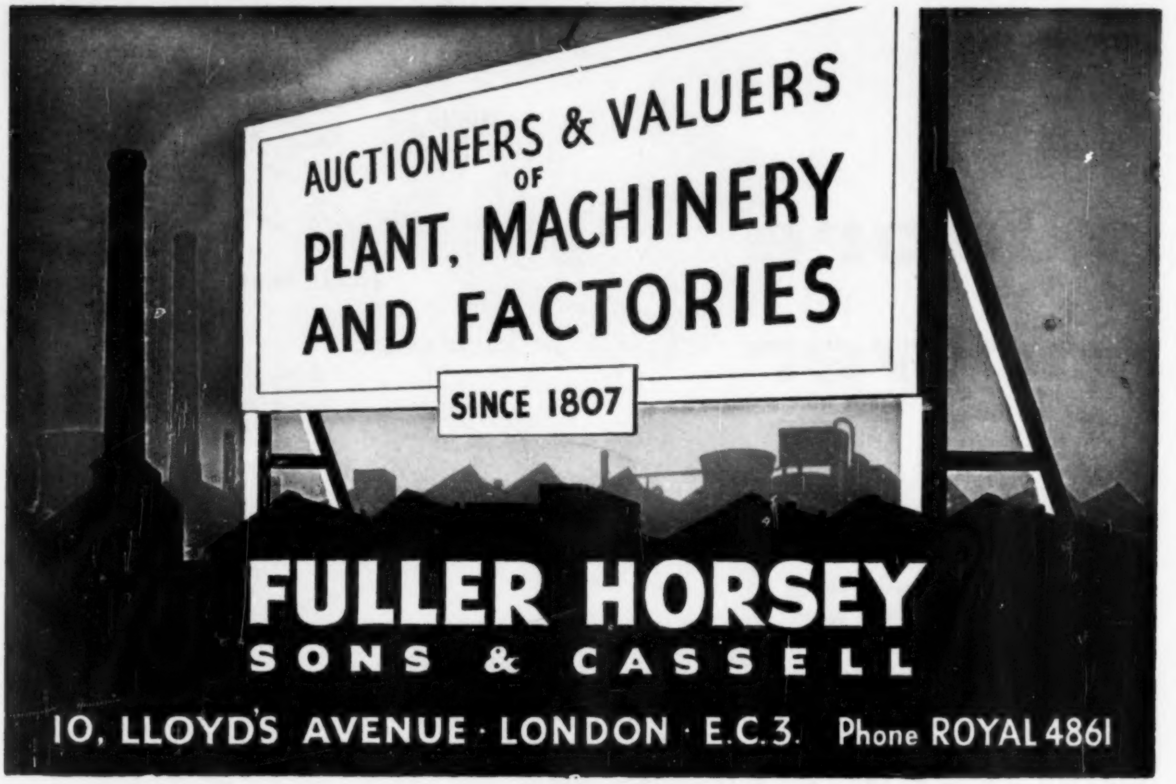

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INDEX TO ADVERTISEMENTS

PAGE
Adamant Engineering Co. Ltd. 15
Aircraft-Marine Products (Gt. Britain) Ltd. 42
Albright & Wilson (Mfg.) Ltd. 61
Aluminium Bronze Co. Ltd. 90
Amal Ltd. 104
Angus, George, & Co. Ltd. 66
Archdale, James, & Co. Ltd. 11
Armstrong Whitworth & Co. (Pneumatic Tools) Ltd. 25
Atlas Copco AB Cover ii
Automotive Engineering Ltd. 111
Automotive Products Co. Ltd. 5, 6, 7, 8

Birfield Industries Ltd. 37, 109
Birmetals Ltd. 50
Booth, James, & Co. Ltd. 103
Borg Warner Ltd. (Morse Chain Div.) 87
B.R.D. Co. Ltd. 92
British Oxygen Co. Ltd., The 28
British Piston Ring Co. Ltd., The 65
British Timken Ltd. Cover i
Brook Motors Ltd. 99
Broom & Wade Ltd. 48
Brown, David, Corporation (Sales) Ltd., The 84, 97
Bury Felt Mfg. Co. 21

Cape Asbestos Co. Ltd., The 70, 71
Cashmore, John, Ltd. Cover iv
Castings Ltd. 12
C.A.V. Ltd. 14
CIBA (A.R.L.) L....
Clancey, G., Ltd. 94
Clayton Dewandre Co. Ltd. 16
Commer Cars Ltd. 114
Concentric Manufacturing Co. Ltd. 46
Coopers Mechanical Joints Ltd. 108
Copper & Asbestos Washer Co. Ltd. 106
Cossor Instruments Ltd. 27
Cross Mfg. Co. (1938) Ltd. 102

Dartmouth Auto Castings Ltd. 74
Delapena & Son Ltd. 80
Deloro Stellite Ltd. 88
Desoutter Bros. Ltd. 56
Doncaster, Daniel, & Sons Ltd. 32
Drummond - Asquith (Sales) Ltd. 79
Dunford & Elliott (Sheffield) Ltd. 108
Dunlop Rubber Co. Ltd. 64, 115

Edwards, F. J., Ltd. 114
Electrical Development Association 23

Fairman Precision Tools & Products Ltd. 110
Farnborough Engineering Co. Ltd. 35
Feeny & Johnson Ltd. 96
Feltham, Walter H., & Sons Ltd. 114
Fenner, J. H., & Co. Ltd. 54
Ferodo Ltd. 49
Firth-Vickers Stainless Steels Ltd. 24
Fletcher Miller Ltd. 92
Flexibox Ltd. 114
Fry's Metal Foundries Ltd. 63
Fuller, Horsey, Sons & Cassell 116

General Electric Co. Ltd., The 69, 86
Girling Ltd. 53, 105

Habershon, J. J., & Sons Ltd. 100
Herbert, Alfred, Ltd. 17, 19
Hermetic Rubber Co. Ltd. 98
Hey Engineering Co. Ltd. 104
Hoffmann Manufacturing Co. Ltd., The 77
Holroyd, John, & Co. Ltd. 95
Holset Engineering Co. Ltd. 81

I.C.I. (General Chemicals, Heat Treatment) Ltd. 52
I.C.I. (Metals) Ltd. 85
I.C.I. (Plastics) Ltd. 78
Ilford Ltd. 40
International Twist Drill Co. Ltd., The 34

Jackson, H., Ltd. 33
Jenks Bros. Ltd. 112

Kayser Ellison & Co. Ltd. 68
Kirkstall Forge Eng. Ltd. 47

Lapointe Machine Tool Co. Ltd., The 4
Laycock Engineering Ltd. 37
Lead Development Association 98
Ley's Malleable Castings Ltd. 41
Long Meadow Felt Co. Ltd. 55

Manganese Bronze & Brass Co. Ltd., The 59
Marsden, Samuel, & Son Ltd. 114
M.C.L. & Repetition Ltd. 114
Mek-Elek Engineering Ltd. 112
Metalastik Ltd. 43
Metropolitan-Vickers Electrical Co. Ltd. 83
Midland Motor Cylinder Co. Ltd., The 36
Moss Gear Co. Ltd. 98

Neill, Jas., & Co. (Sheffield) Ltd. 82
Newall, A. P., & Co. Ltd. 20

Park Gate Iron & Steel Co. Ltd., The 1
Philidas Div. of Whitehouse Industries Ltd. 26
Pollard Bearings Ltd. 60
Power Jacks Ltd. 102

Ransome & Marles Bearing Co. Ltd. 73
Renold Chains Ltd. 45
Rockwell Machine Tool Co. Ltd. 22

S. & D. Rivet Co. 110
Salter, Geo., & Co. Ltd. 10, 107
Sandwell Casting Co. 112
Sankey Green Wire Weaving Co. Ltd. 80
Schrader's, A., Son 67
Sheffield Twist Drill & Steel Co. Ltd., The 13, 51
Shotton Bros Ltd. Cover iii
Silentbloc Ltd. 57
Simmonds Aerocessories Ltd. 39
Simms Motor Units Ltd. 2
Sintered Products Ltd. 84
Skefko Ball Bearing Co. Ltd. 76
Small & Parkes Ltd. 106
Speed Tools Ltd. 94
Steel Co. of Wales Ltd., The 18
Steel, J. M., & Co. Ltd. 90
Sterling Metals Ltd. 72
Sutcliffe Speakman & Co. Ltd. 104
Super Oil Seals & Gaskets Ltd. 30
Sykes, W. E., Ltd. 31

Terry, Herbert, & Sons Ltd. 96, 101
Thompson, John, Motor Pressings Ltd. 89
Tilghman's Ltd. 82
Turner Bros. Asbestos Co. Ltd. 110
Turner Machine Tools Ltd. 102

Udal, J. P., Ltd. 100

Van Moppes, L. M., & Sons (Diamond Tools) Ltd. 9
Vaughan, A. J., & Co. (Mitre Works) Ltd. 108
Vaughan Associates Ltd. 3

Ward, Thos. W., Ltd. 38
Webb Conveyors & Automation Ltd. 29
Wellworthy Piston Rings Ltd. 62
Weston, Charles, & Co. Ltd. 58
West Yorkshire Foundries Ltd. 44
Wild-Barfield Electric Furnaces Ltd 113
Wiley, James, & Son Ltd. 86
Wilmot Breeden 91

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